Hydrodynamic fluid film bearings and their effect on the stability of rotating machinery

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Lubricated Journal Bearings

Radial and axial load support of rotating machinery – low friction and long life

Advantages

Do not require external source of pressure.

Support heavy loads. The load support is a function of the lubricant viscosity, surface speed, surface area, film thickness and geometry of the bearing.

Long life (infinite in theory) without wear of surfaces.

Provide stiffness and damping coefficients of large magnitude.

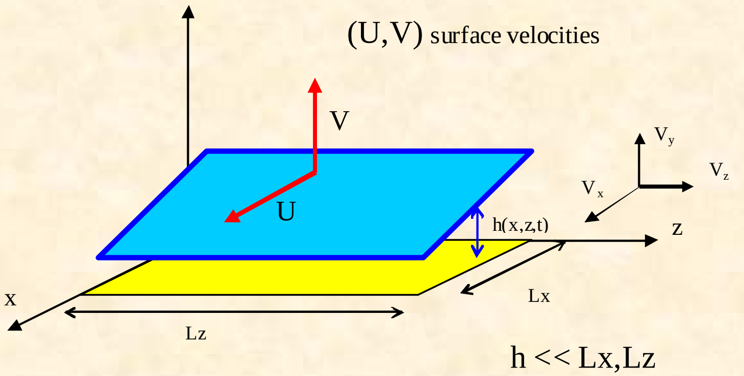
Disadvantages

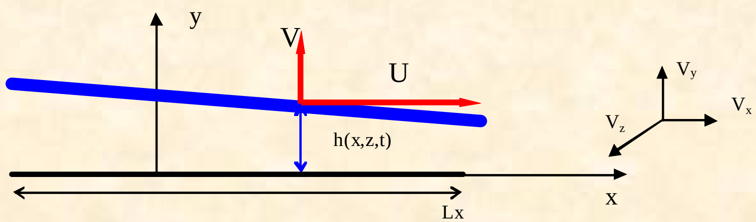
Thermal effects affect performance if film thickness is too small or available flow rate is too low.

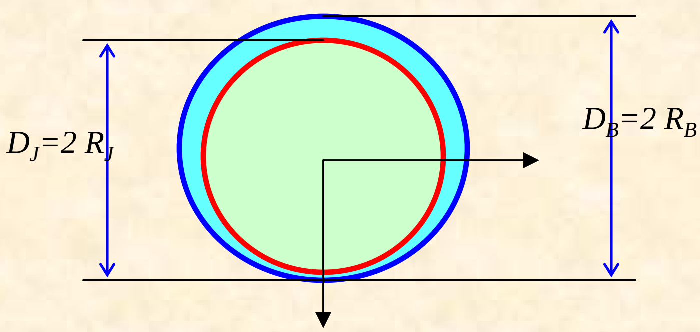
Potential to induce hydrodynamic instability, i.e. loss of effective damping for operation well above critical speed of rotor-bearing system

Typically use MINERAL OIL as lubricant. Modern trend is to replace with working fluid (water)

Fundamentals of Thin Film Lubrication







Cylindrical bearing

- **Film thickness << other dimensions**
- **No curvature effects**
- **Laminar flow, inertialess**

TYP (c/L^*) = 0.001

$$Re = \frac{\rho U_* c}{\mu}$$

SMALL Couette flow Reynolds #

<u>Flow equations:</u> continuity + momentum (x,y)

$$\frac{\partial (v_x)}{\partial x} + \frac{\partial (v_y)}{\partial y} + \frac{\partial (v_z)}{\partial z} = 0$$

$$0 = -\frac{\partial P}{\partial x} + \mu \frac{\partial^2 v_x}{\partial y^2}; \quad 0 = -\frac{\partial P}{\partial z} + \mu \frac{\partial^2 v_x}{\partial y^2}$$

Quasi-static (pressure forces = viscous forces)

Figures 1 & 2 **Geometry of flow region in a thin fluid film bearing (h << Lx, Lz)**

Importance of fluid inertia in thin film flows

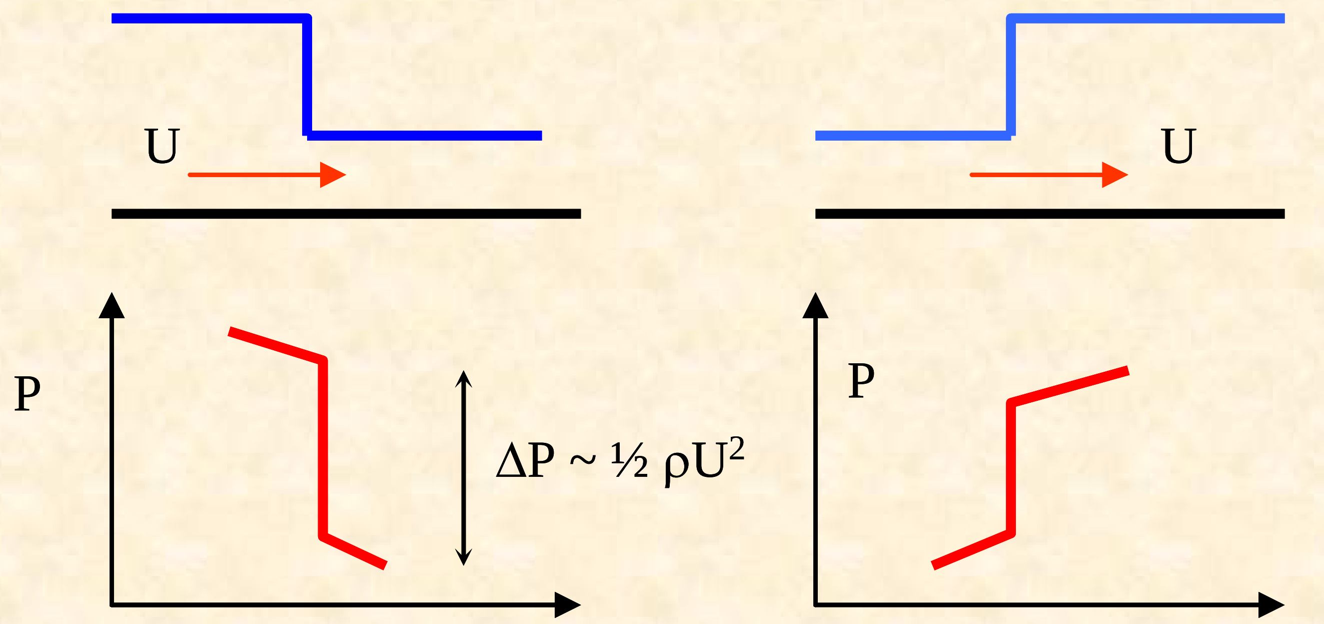
Reynolds numbers

fluid	Absolute viscosity (μ) lbm.ft.s x 10^{-5}	Kinematic viscosity (ν) centistoke	**Re** at 1,000 rpm	**Re at 10,000 rpm**
Air	1.23	15.4	9.9	99
Thick oil	**1,682**	**30.0**	**5.1**	**51**
Light oil	120	2.14	71	711
Water	**64**	**1.00**	**159**	**1,588**
Liquid hydrogen	1.075	0.216	705	**7,052**
Liquid oxygen	10.47	0.191	794	**7,942**
Liquid nitrogen	13.93	0.179	848	**8,477**
R134 refrigerant	13.30	0.163	930	**9,296**

Fluid inertia is important for operation at high speeds and with process fluids. These are prevalent conditions in HP turbomachinery

Table 1 **Importance of fluid inertia effects on several fluid film bearing applications. (c/R_J)=0.001, R_J =38.1 mm (1.5 inch)**

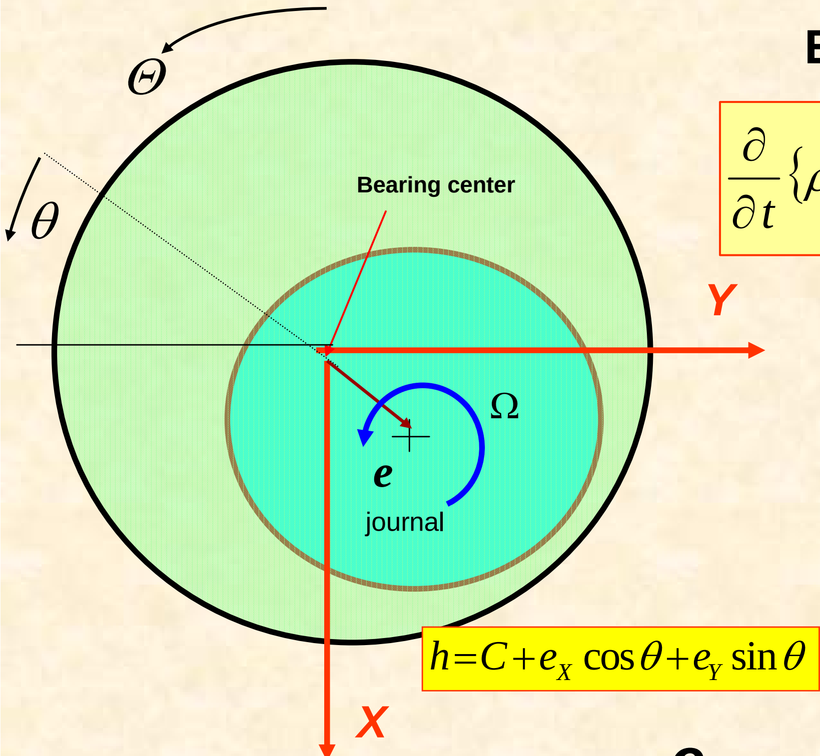
Fluid inertia effects at inlet & edges



Fluid inertia (Bernoulli's effect) causes sudden pressure drop (or raise) at sharp inlets (exits). Most important effect on annular pressure seals and hydrostatic bearings with process fluids

Figure 3 **Pressure drop & rise at sudden changes in film thickness**

Thin Film Lubrication: Reynolds Equation



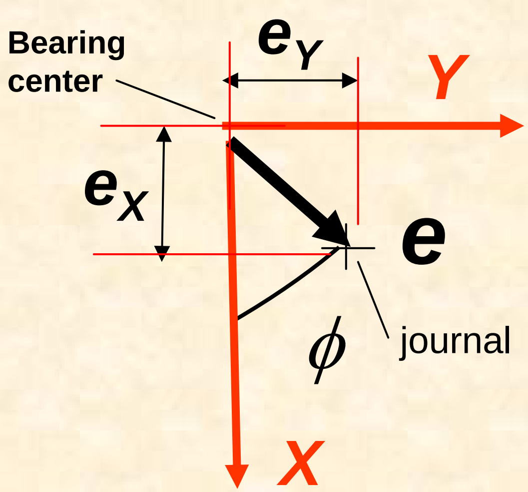
Elliptical PDE in film region

$$\frac{\partial}{\partial t}\{\rho h\} + \frac{\Omega}{2}\frac{\partial}{\partial \Theta}\{\rho\, h\} = \frac{1}{R^2}\frac{\partial}{\partial \Theta}\left\{\frac{\rho\, h^3}{12\,\mu}\frac{\partial P}{\partial \Theta}\right\} + \frac{\partial}{\partial z}\left\{\frac{\rho\, h^3}{12\,\mu}\frac{\partial P}{\partial z}\right\}$$

Film thickness

$$h = c + e_X \cos\Theta + e_Y \sin\Theta = e \sin\theta$$

Pressure = ambient on sides
Pressure > $P_{\text{cavitation}}$

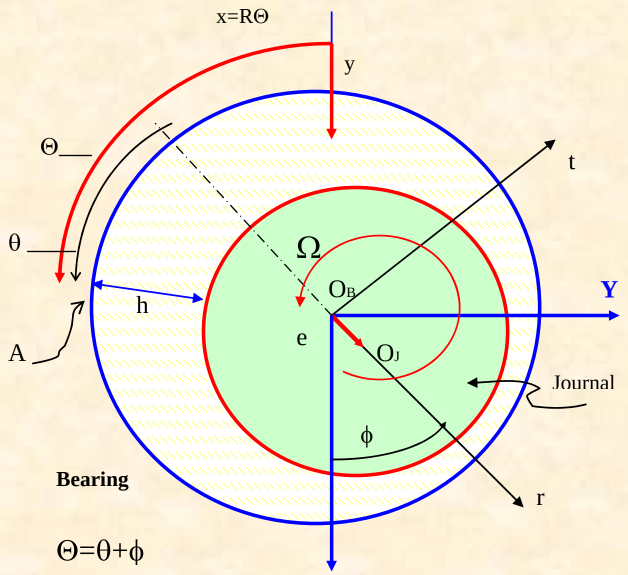


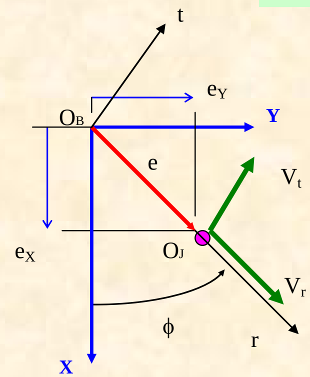
<u>Kinematics of journal motion:</u>

$e_X = e \cos(\phi)$; $e_Y = e \sin(\phi)$

Figure 4 **Cylindrical journal bearing & coordinates**

Kinematics of journal motion





$e_X = e\cos(\phi);\quad e_Y = e\sin(\phi)$

$$\begin{bmatrix}\dot{e}_X\\ \dot{e}_Y\end{bmatrix}=\begin{bmatrix}\cos\phi & -\sin\phi\\ \sin\phi & \cos\phi\end{bmatrix}\begin{bmatrix}\dot{e}\\ e\dot{\phi}\end{bmatrix}$$

Set: incompressible fluid (oil)

Reynolds Eqn. in fixed coordinates (X,Y)

$$\frac{1}{R^2}\frac{\partial}{\partial\Theta}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial\Theta}\right\}+\frac{\partial}{\partial z}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial z}\right\}=\left\{\dot{e}_X+e_Y\frac{\Omega}{2}\right\}\cos\Theta+\left\{\dot{e}_Y-\frac{\Omega}{2}e_X\right\}\sin\Theta$$

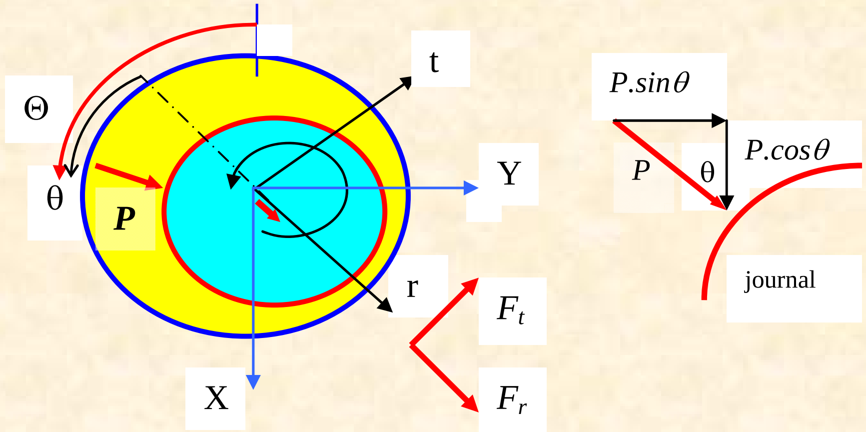
Reynolds Eqn. in moving coordinates)

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial\theta}\right\}+\frac{\partial}{\partial z}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial z}\right\}=\dot{e}\cos\theta+e\left\{\dot{\phi}-\frac{\Omega}{2}\right\}\sin\theta$$

For circular centered orbits:: radius (e) and $\dot{\phi}=\Omega/2$

Loss of load capacity ⇨ **Hydrodynamic pressure *P=0***

Journal bearing reaction force



Force = integration of pressure field on journal surface

$$\begin{bmatrix} F_r \\ F_t \end{bmatrix} = \int_0^L \int_0^{2\pi} P(\theta, z, t) \begin{bmatrix} \cos\theta \\ \sin\theta \end{bmatrix} R \cdot d\theta \, dz$$

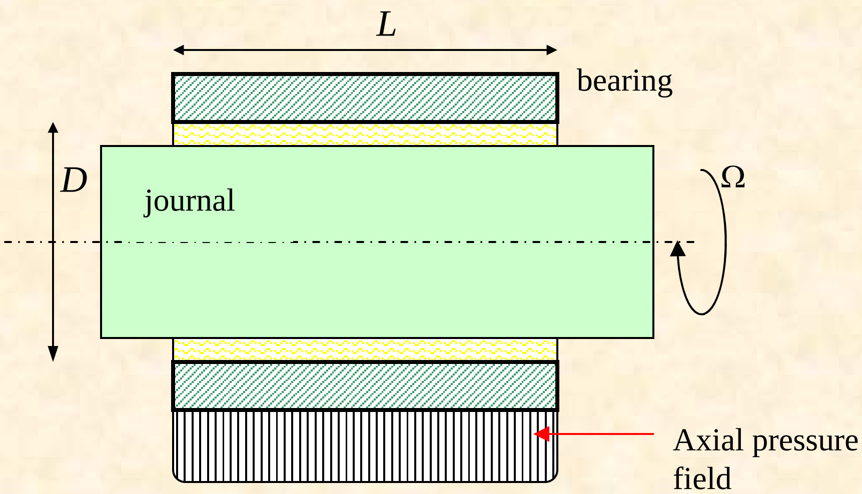
$$\begin{bmatrix} F_X \\ F_Y \end{bmatrix} = \begin{bmatrix} \cos\phi & -\sin\phi \\ \sin\phi & \cos\phi \end{bmatrix} \begin{bmatrix} F_r \\ F_t \end{bmatrix}$$

Dynamic forces = fn. of journal position and velocities, rotational speed (Ω), viscosity (μ) and geometry (*L, D, c*)

$$F_\alpha = F_\alpha\left(\Omega, \dot{e}_X, \dot{e}_Y\right) = F_\alpha\left(\dot{e}, e\left[\dot{\phi} - \frac{\Omega}{2}\right]\right)$$

Figure 5 **Fluid film force acting on journal surface**

LONG journal bearing (limit geometry)



LONG BEARING MODEL

***L/D* >> 1**

***dP/dz* $\rightarrow$ 0**

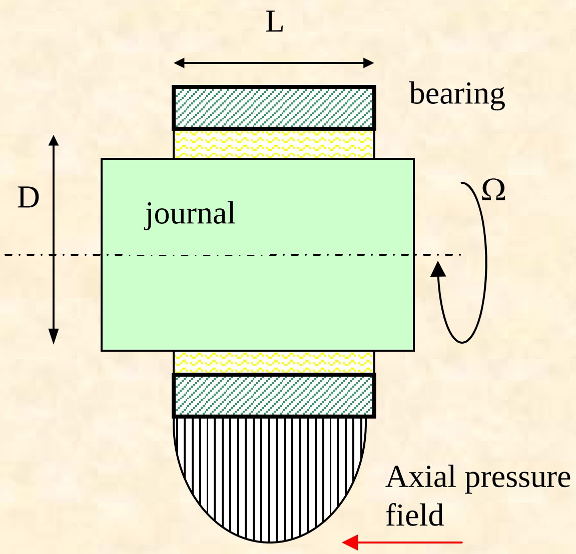
***L/D* >>> 1**

$$\frac{\partial}{\partial t}\{h\}+\frac{\Omega}{2}\frac{\partial}{\partial \Theta}\{h\}=\frac{\partial}{\partial z}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial z}\right\}$$

Pressure does not vary axially. Not applicable for most practical cases, except sealed squeeze film dampers

Figure 6

SHORT journal bearing (limit geometry)



***L/D* << 1**

***dP/dθ* → 0**

***L/D* < 0.50**

$$\frac{1}{R^2}\frac{\partial}{\partial\theta}\left\{\frac{h^3}{12\mu}\frac{\partial P}{\partial\theta}\right\}=\frac{\partial}{\partial t}\{h\}+\frac{\Omega}{2}\frac{\partial}{\partial\Theta}\{h\}$$

Applicable to actual rotating machinery

SHORT JOURNAL BEARING MODEL

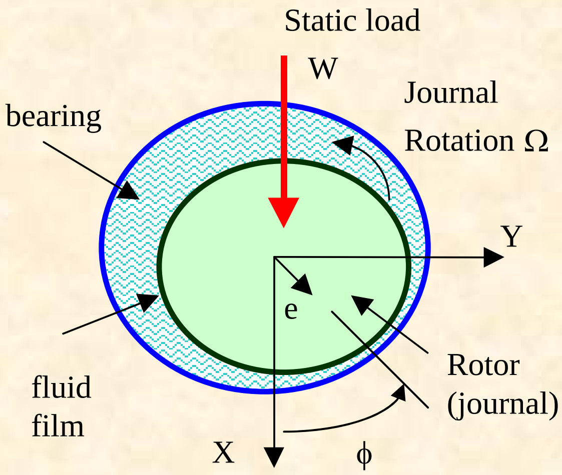
$$P(\theta,z,t)-P_a=\frac{6\mu\left[\dot e\cos\theta+e\left(\dot\phi-\frac{\Omega}{2}\right)\sin\theta\right]}{C^3 \qquad H^3}\left\{z^2-\left(\frac{L}{2}\right)^2\right\}$$

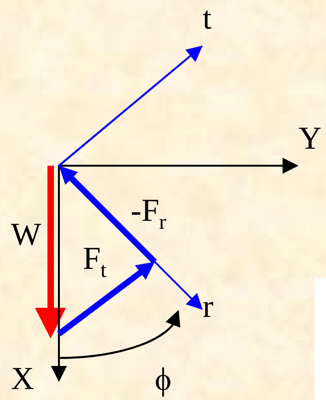
Hydrodynamic pressure is proportional to viscosity (μ), speed (Ω), and most important to: $1/C^3$

Control of tolerances in machined clearance is critical for reliable performance

Figure 7

STATIC LOAD PERFORMANCE



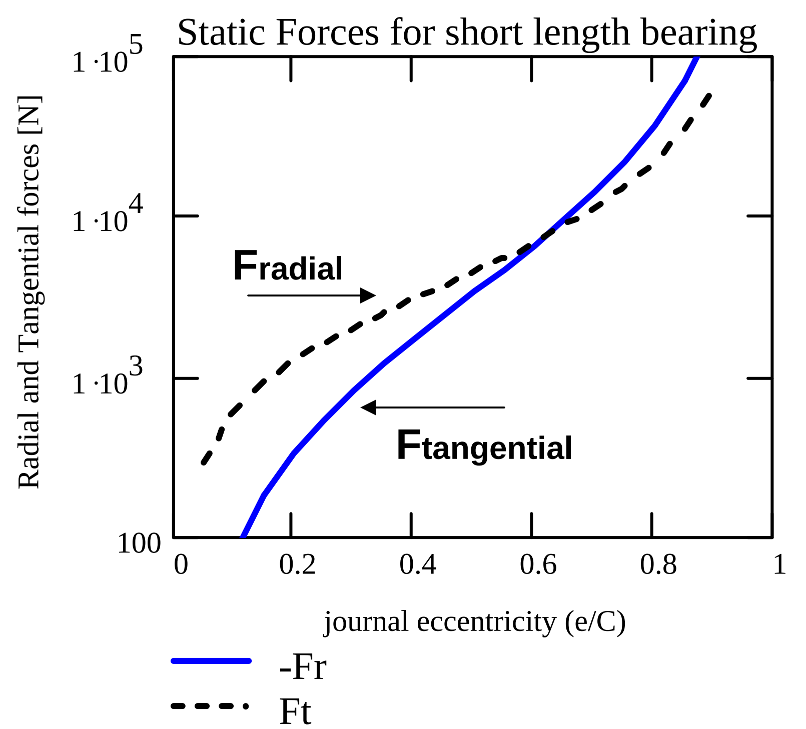


Bearing reaction force = applied static load (% of rotor weight)

$$F_r = -\frac{\mu R L^3 \Omega}{c^3} \frac{\varepsilon^2}{\left(1-\varepsilon^2\right)^2}; \quad F_t = +\frac{\mu R L^3 \Omega}{c^2} \frac{\pi \cdot \varepsilon}{4\left(1-\varepsilon^2\right)^{3/2}}$$

Force Balance for Static Load

Radial and tangential forces for *L/D*=0.25 bearing. μ=0.019 Pa.s, L=0.05 m, c=0.1 mm, 3, 000 rpm,



Journal bearing can generate large reaction forces. Highly nonlinear functions of journal eccentricity

*

Figures 8 & 9

DESIGN PARAMETER: STATIC LOAD PERFORMANCE

Sommerfeld number

$$S = \frac{\mu \; N \, L \, D}{W}\left(\frac{R}{c}\right)^2$$

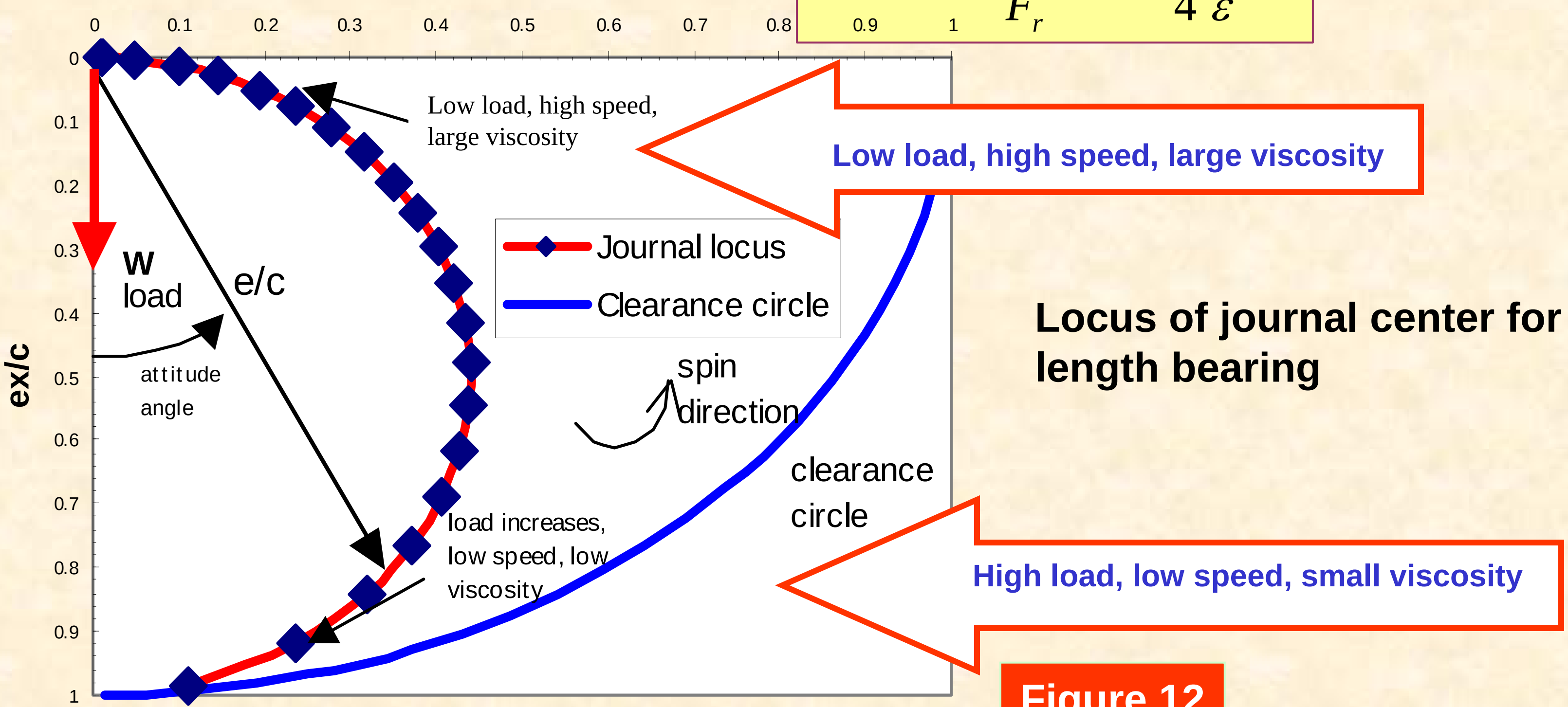
N **rotational speed (rev/s)**
W **static load**
***L*, *D*=2*R*, *c* : clearance &**
μ viscosity

Given *S*, iterative solution to find operating journal eccentricity (ε = *e/c*) and attitude angle (ϕ):

$$\sigma = \pi \, S \, (L/D)^2 = \frac{\mu \, \Omega \, L \, R}{4W}\left(\frac{L}{c}\right)^2 = \frac{\left(1-\varepsilon^2\right)^2}{\varepsilon \sqrt{\left\{16\varepsilon^2 + \pi^2\left(1-\varepsilon^2\right)\right\}}}$$

$$\tan\phi = -\frac{F_t}{F_r} = \frac{\pi\sqrt{\left(1-\varepsilon^2\right)}}{4\,\varepsilon}$$

Attitude angle



Locus of journal center for short length bearing

Figure 12

DESIGN PARAMETER: STATIC LOAD PERFORMANCE

Sommerfeld number

$$\sigma = \pi\, S\,(L/D)^2 = \frac{\mu\,\Omega\, L\, R}{4W}\left(\frac{L}{c}\right)^2$$

N **rotational speed (rev/s)**
W **static load**
***L*, *D*=2*R*, *c* : clearance &**
μ **viscosity**

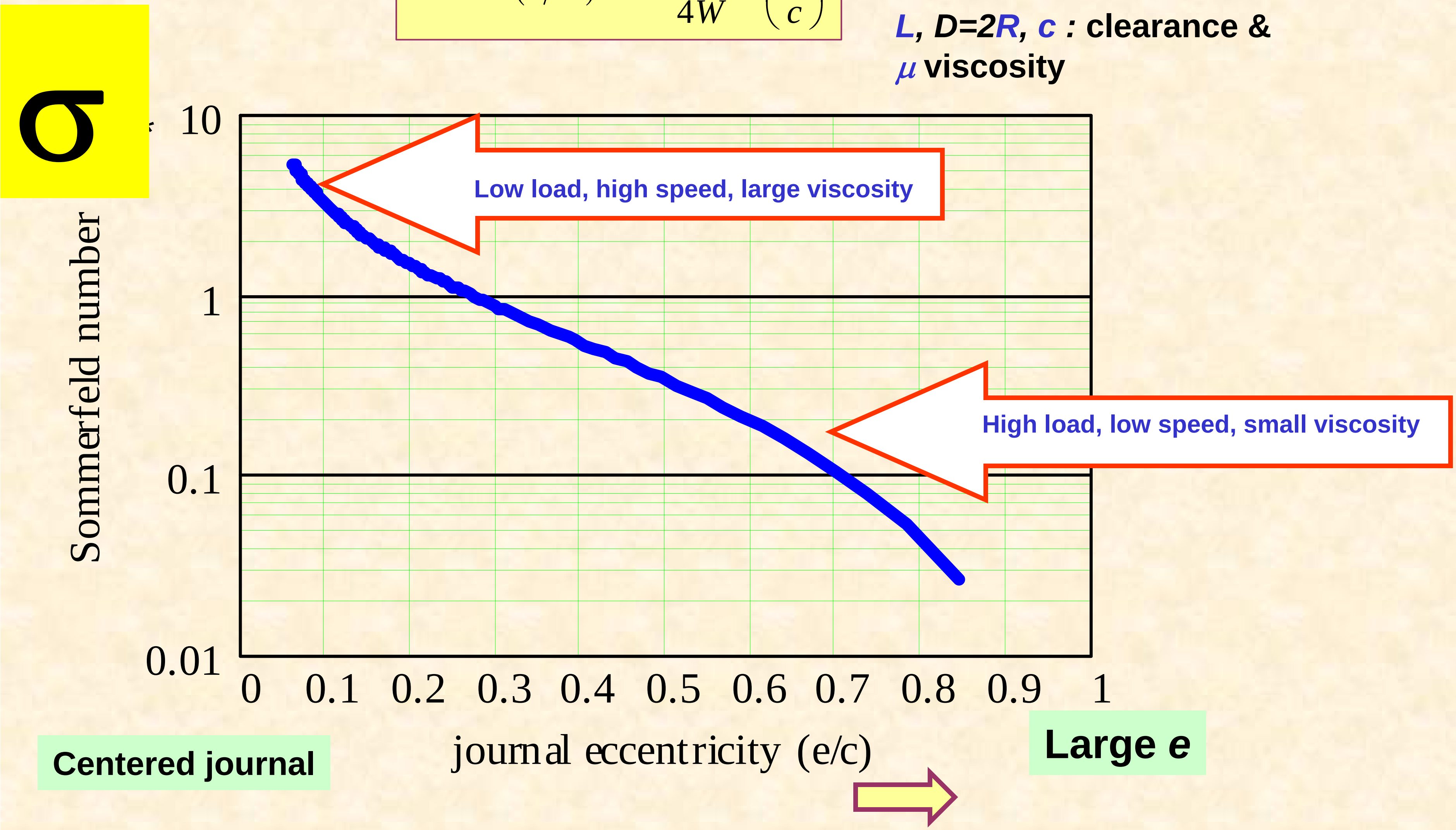


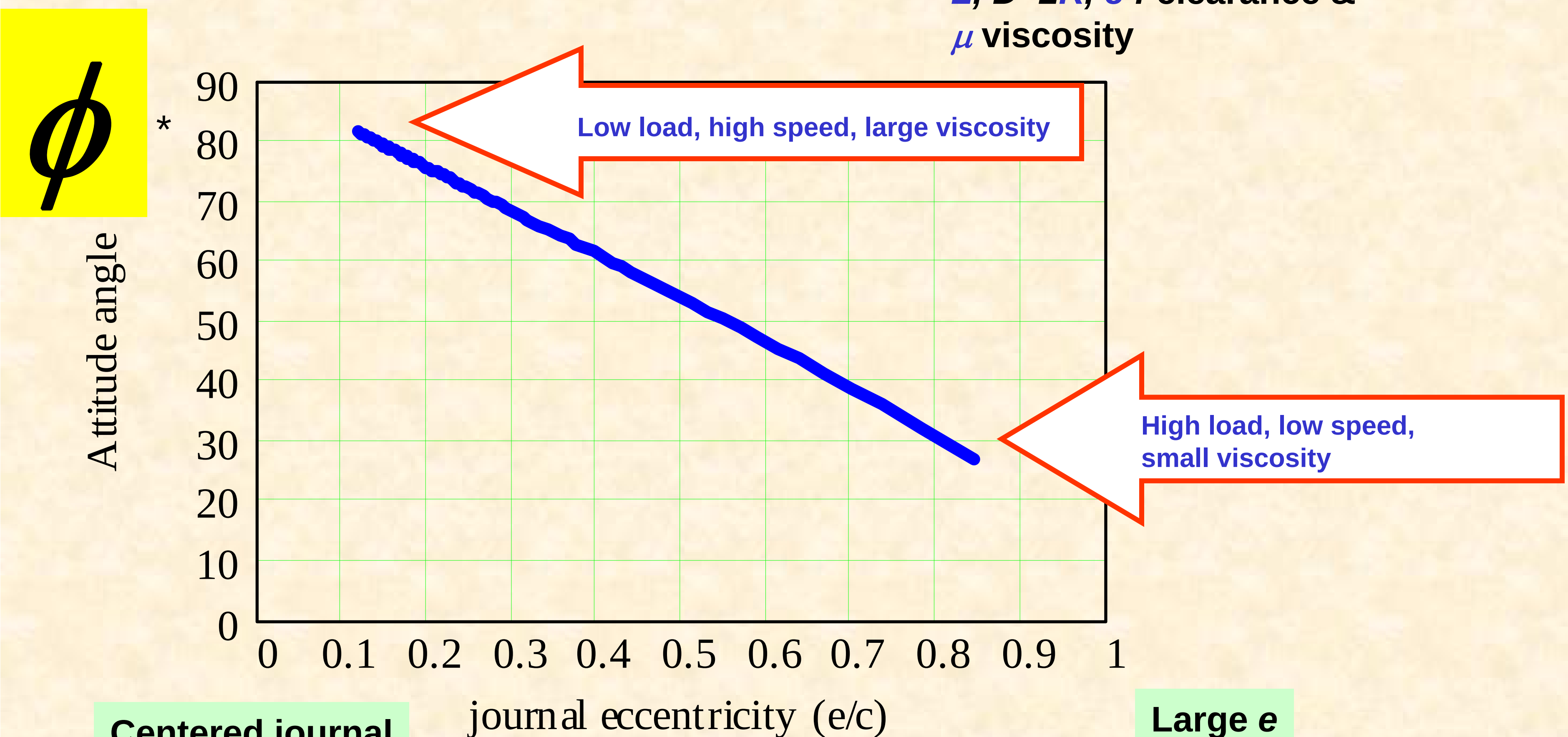
Figure 10 **Sommerfeld # vs journal eccentricity**

DESIGN PARAMETER: STATIC LOAD PERFORMANCE

Sommerfeld number

$$\sigma = \pi S \left(L/D\right)^2 = \frac{\mu\, \Omega\, L\, R}{4W}\left(\frac{L}{c}\right)^2$$

N **rotational speed (rev/s)**
W **static load**
L**,** ***D*****=2*****R*****,** ***c*** **: clearance &**
μ **viscosity**



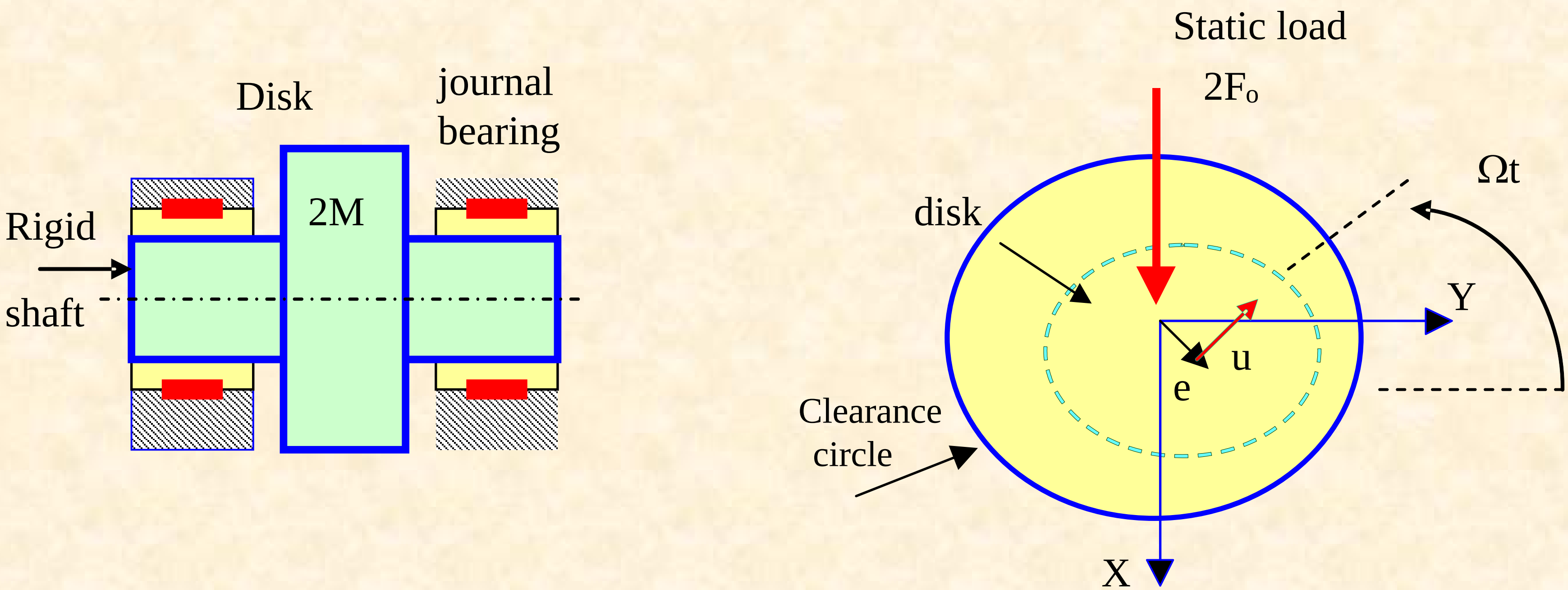
Centered journal

Large *e*

Figure 11 **Attitude angle # vs journal eccentricity**

DYNAMICS OF ROTOR-BEARING SYSTEM

Symmetric - rigid rotor supported on short length journal bearings



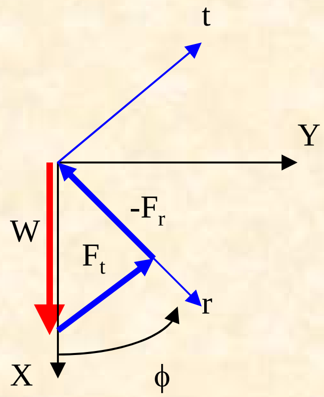
Equations of motion:

$$M\,\ddot{X} = F_X + M\,u\,\Omega^2 \sin(\Omega\,t) + F_o$$

$$M\,\ddot{Y} = F_Y + M\,u\,\Omega^2 \cos(\Omega\,t)$$

Figure 13 **Rigid rotor supported on journal bearings.**
(u) imbalance, (e) journal eccentricity

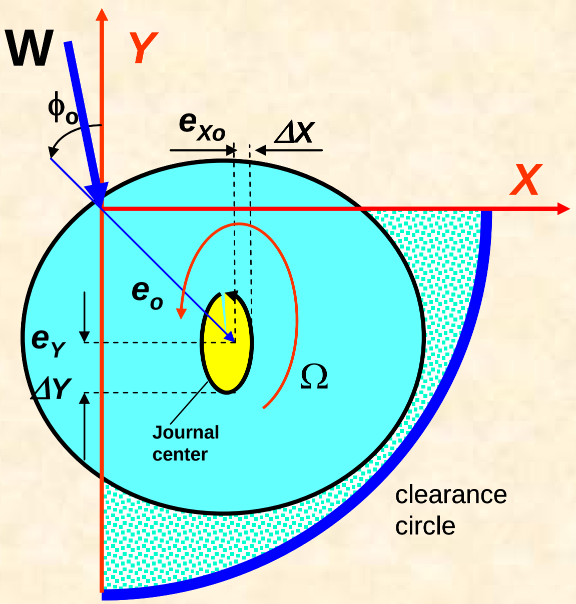
Consider small amplitude motions about static equilibrium position (SEP). SEP defined by applied static load.



$$F_{X_O} = -F_o, \quad F_{Y_O} = 0, \quad \Rightarrow e_{X_O}, e_{Y_O} \text{ or } e_O, \phi_O$$

Let: $$e_X = e_{X_O} + \Delta e_X(t), \quad e_Y = e_{Y_O} + \Delta e_Y(t)$$

Static load



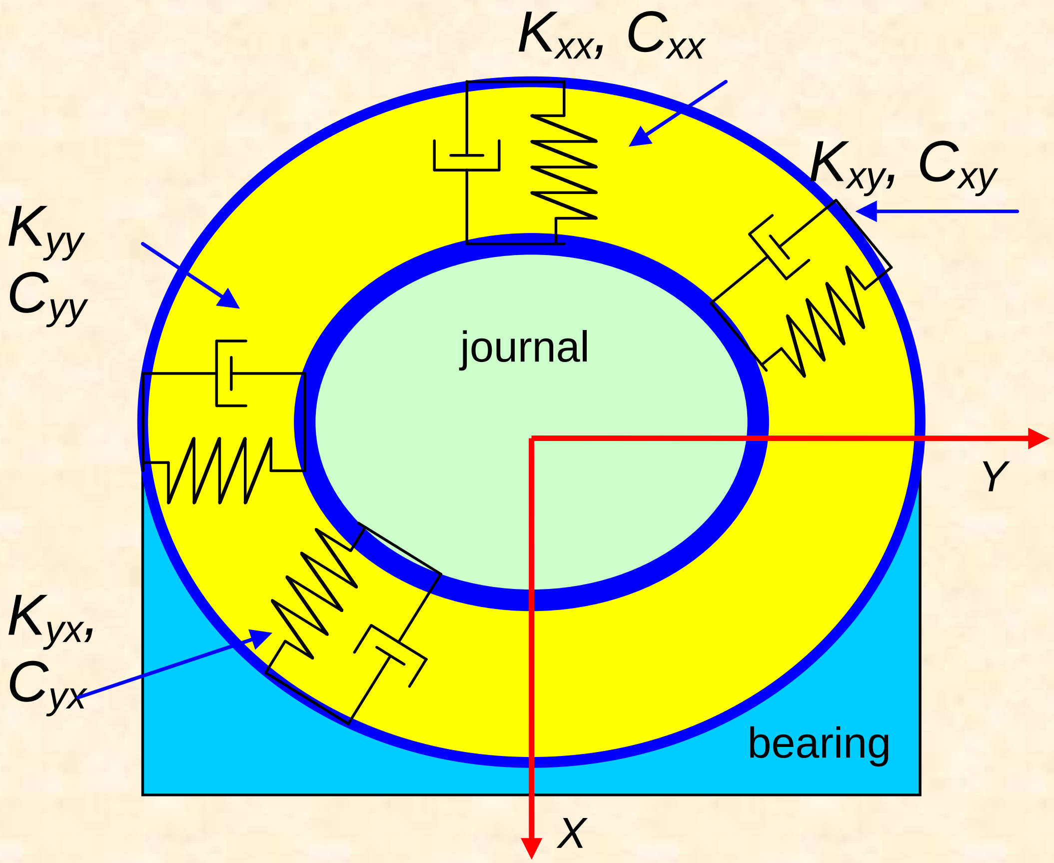
Expansion of forces abut SEP

$$F_X = F_{X_O} + \frac{\partial F_X}{\partial X}\Delta X + \frac{\partial F_X}{\partial Y}\Delta Y + \frac{\partial F_X}{\partial \dot{X}}\Delta \dot{X} + \frac{\partial F_X}{\partial \dot{Y}}\Delta \dot{Y}$$

$$F_Y = F_{Y_O} + \frac{\partial F_Y}{\partial X}\Delta X + \frac{\partial F_Y}{\partial Y}\Delta Y + \frac{\partial F_Y}{\partial \dot{X}}\Delta \dot{X} + \frac{\partial F_Y}{\partial \dot{Y}}\Delta \dot{Y}$$

Figure 14 **Small amplitude journal motions about an equilibrium position**

ROTORDYNAMIC FORCE COEFFICIENTS



The "physical representation" of stiffness and damping coefficients in lubricated bearings

Stiffness: $K_{ij} = -\dfrac{\partial F_i}{\partial X_j}$;

Damping: $C_{ij} = -\dfrac{\partial F_i}{\partial \dot{X}_j}$

Inertia: $M_{ij} = -\dfrac{\partial F_i}{\partial \ddot{X}_j}$;

i,j = X,Y

Strictly valid for small amplitude motions. Derived from SEP

Figure 15

ROTORDYNAMIC FORCE COEFFICIENTS

Static reaction force:

Stiffness Matrix:

Damping Matrix:

$$\begin{pmatrix} F_X(t) \\ F_Y(t) \end{pmatrix} = \begin{bmatrix} F_{X_O} \\ F_{Y_O} \end{bmatrix} - \begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix} \begin{pmatrix} \Delta X \\ \Delta Y \end{pmatrix} - \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \begin{pmatrix} \Delta \dot{X} \\ \Delta \dot{Y} \end{pmatrix}$$

Inertia ~ 0 in journal bearings

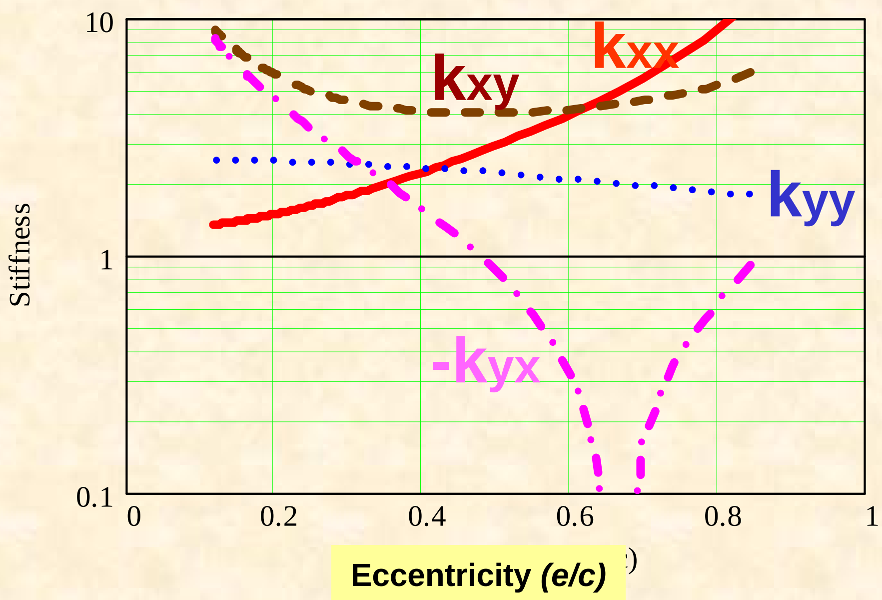
Linearized Equations of motion

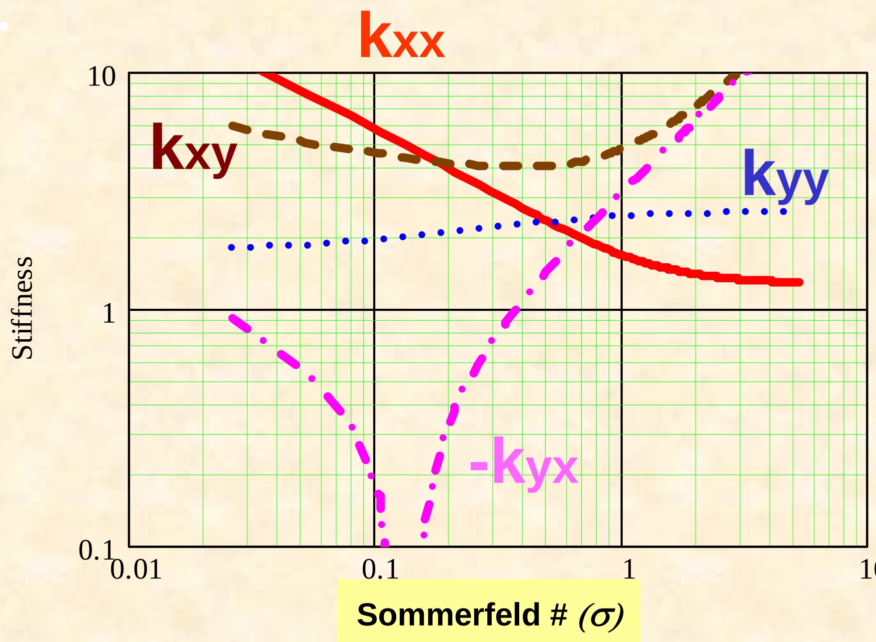
$$\begin{bmatrix} M & O \\ O & M \end{bmatrix} \begin{pmatrix} \Delta \ddot{X} \\ \Delta \ddot{Y} \end{pmatrix} + \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix} \begin{pmatrix} \Delta \dot{X} \\ \Delta \dot{Y} \end{pmatrix} + \begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix} \begin{pmatrix} \Delta X \\ \Delta Y \end{pmatrix} = M\, u\, \Omega^2 \begin{pmatrix} \cos \Omega t \\ \sin \Omega t \end{pmatrix}$$

Strictly valid for small amplitude motions. Derived from SEP

Journal Bearing: STIFFNESS COEFFICIENTS

$$\sigma = \frac{\mu\,\Omega\,L\,R}{4W}\left(\frac{L}{c}\right)^2$$





High speed
Low load
Large viscosity

Low speed
Large load
Low viscosity

High speed
Low load
Large viscosity

$$\sigma = \frac{\mu\,\Omega\,L\,R}{4W}\left(\frac{L}{c}\right)^2$$

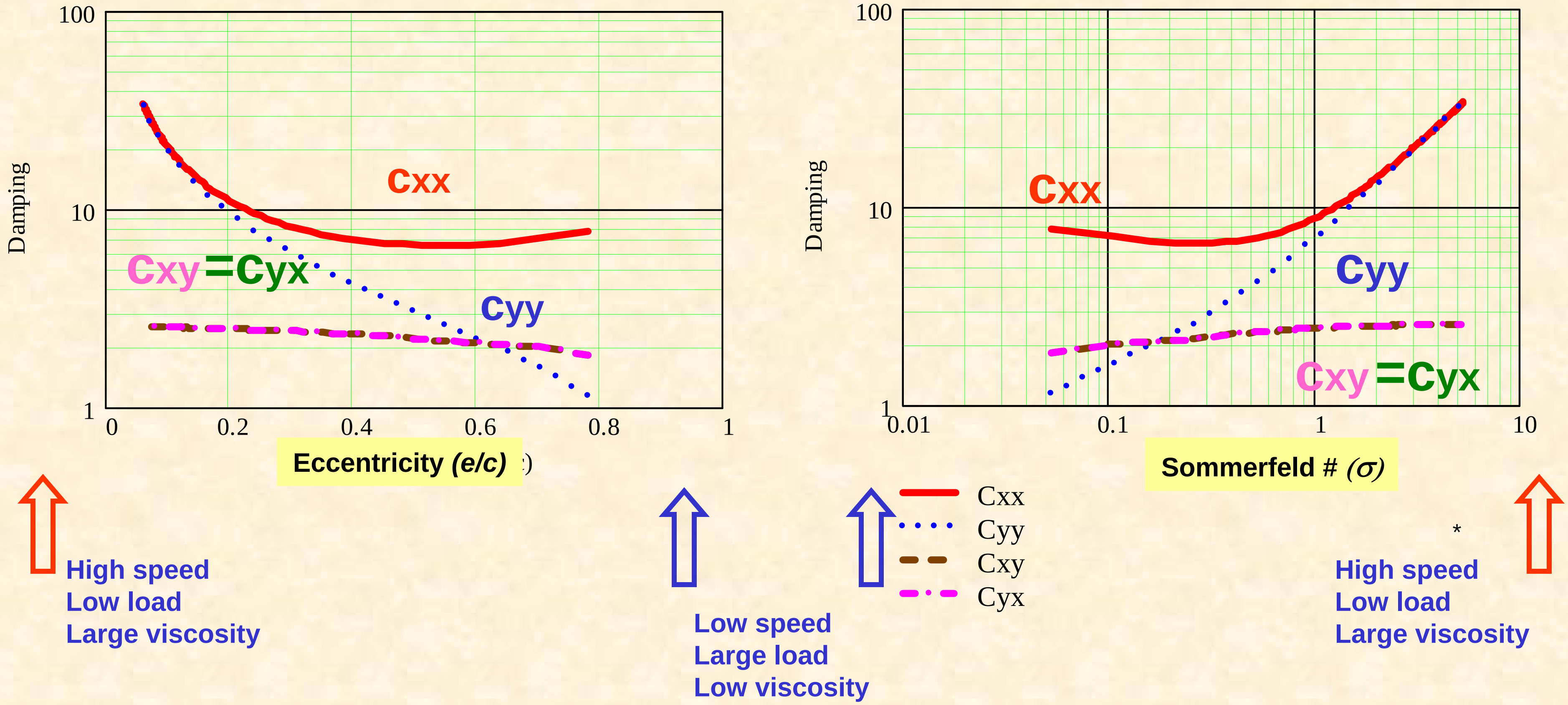
$$k_{\alpha\beta} = K_{\alpha\beta}\,(c/F_o)$$

Care with non dimensional value interpretation

Figure 16 & 17 Bearing stiffnesses versus eccentricity and design number (σ)

Journal Bearing: DAMPING COEFFICIENTS

$$\sigma = \frac{\mu \,\Omega\, L\, R}{4W}\left(\frac{L}{c}\right)^2$$



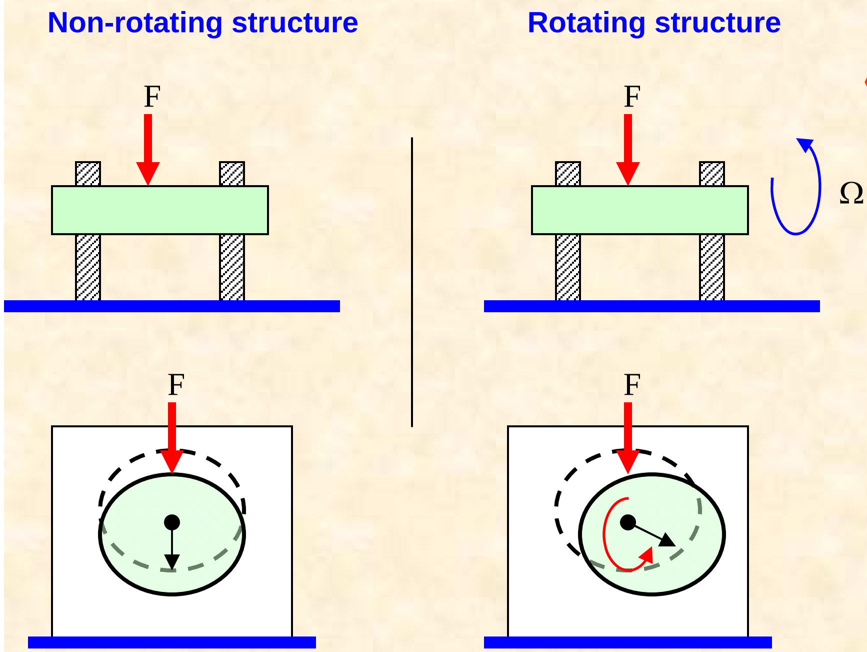
$$c_{\alpha\beta} = C_{\alpha\beta}\ (c\Omega/F_o)$$

Care with non dimensional value interpretation

$$\sigma = \frac{\mu \,\Omega\, L\, R}{4W}\left(\frac{L}{c}\right)^2$$

Figure 16 & 17 Bearing damping versus eccentricity and design number (σ)

Journal Bearing: OPERATION at CENTERED CONDITION



High speed
Low load
Large viscosity

$$e_o \rightarrow 0, \ \phi_o = 90 \text{ deg}$$

$$K_{xx} = K_{yy} = 0$$

no direct stiffness

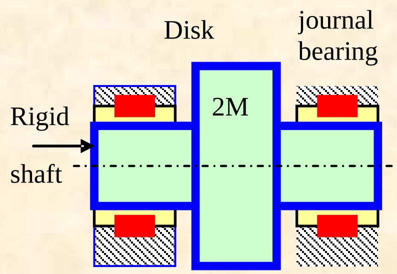
Significance of cross-coupled effect in journal bearing

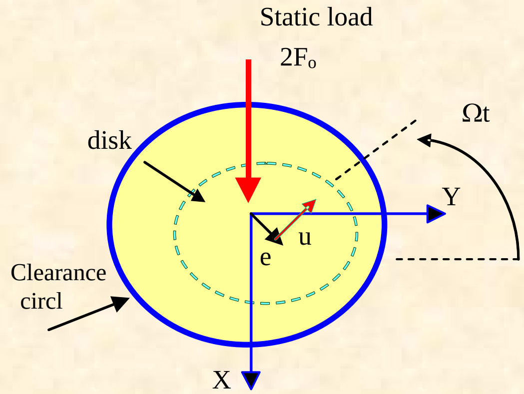
$$K_{xy} = C_{xx}\ \Omega/2$$

$$K_{XY} = -K_{YX} = \bar{k} = \frac{\mu \Omega R L^3}{c^3}\frac{\pi}{4} = \frac{\Omega}{2}\bar{c}; \qquad C_{XX} = C_{YY} = \bar{c} = \frac{\mu R L^3}{c^3}\frac{\pi}{2}$$

Pure cross-coupling effect

STABILITY OF ROTOR-BEARING SYSTEM



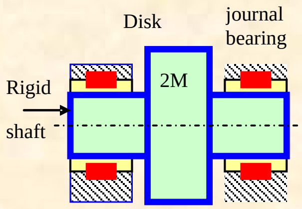


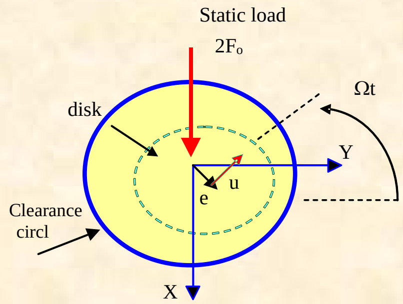
$$\begin{bmatrix} M & O \\ O & M \end{bmatrix}\begin{pmatrix} \Delta\ddot{X} \\ \Delta\ddot{Y} \end{pmatrix} + \begin{bmatrix} C_{XX} & C_{XY} \\ C_{YX} & C_{YY} \end{bmatrix}\begin{pmatrix} \Delta\dot{X} \\ \Delta\dot{Y} \end{pmatrix} + \begin{bmatrix} K_{XX} & K_{XY} \\ K_{YX} & K_{YY} \end{bmatrix}\begin{pmatrix} \Delta X \\ \Delta Y \end{pmatrix} = \begin{pmatrix} 0 \\ 0 \end{pmatrix}$$

If rotor-bearing system is to become unstable, this will occur at a <u>threshold speed of rotation</u> (Ωs) with rotor performing (undamped) orbital motions at a <u>whirl frequency</u> (ωs)

$$x = Ae^{j\omega_s t} = Ae^{j\bar{\omega}\tau} \; ; \; y = B\,e^{j\omega_s t} = Be^{j\bar{\omega}\tau}; \quad j = \sqrt{-1}$$

STABILITY OF ROTOR-BEARING SYSTEM





Equivalent support stiffness

$$p_s^2\overline{\omega}_s^2 = k_{eq} = \frac{k_{XX}c_{YY} + k_{YY}c_{XX} - c_{YX}k_{XY} - c_{XY}k_{YX}}{c_{XX} + c_{YY}} = \frac{C\,M\omega_S^2}{F_o}$$

Whirl frequency ratio

$$\overline{\omega}_s^2 = \frac{\left(k_{eq} - k_{XX}\right)\left(k_{eq} - k_{YY}\right) - k_{XY}\cdot k_{YX}}{c_{XX}c_{YY} - c_{XY}c_{YX}} = \left(\frac{\omega_s}{\Omega_s}\right)^2$$

= whirl frequency (ωs)/threshold speed instability (Ωs)

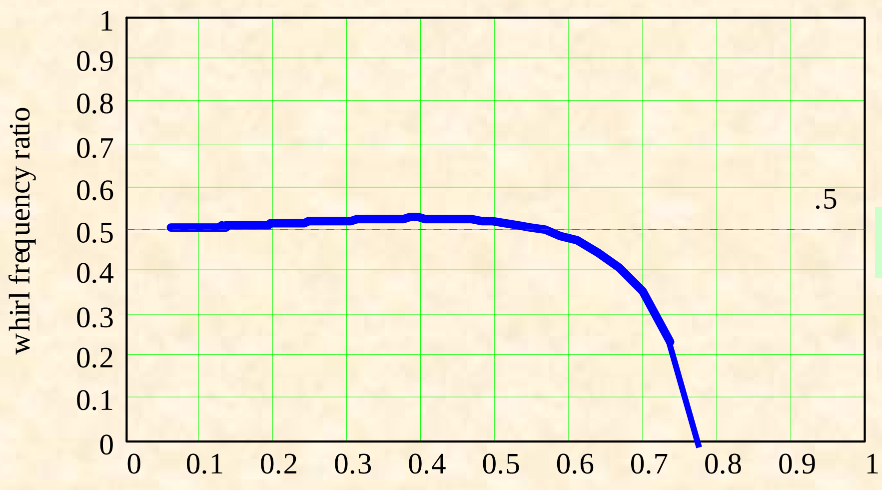
The *WFR* is independent of the rotor characteristics (rotor mass and flexibility)

$$M\,\omega_s^2 = k_{eq}\left(\frac{F_o}{C}\right) = K_{eq} \;\Rightarrow\; \omega_s = \sqrt{\frac{K_{eq}}{M}} = \omega_n \;\Rightarrow$$

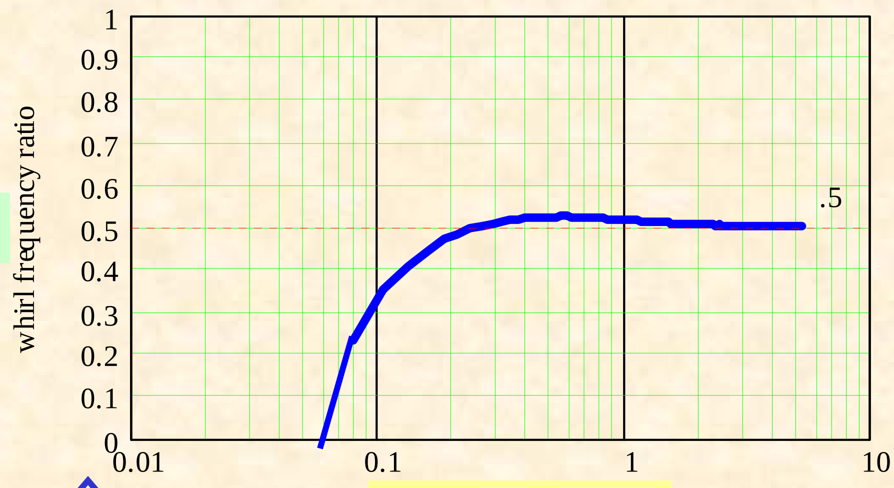
whirl frequency equals the natural frequency of rigid rotor supported on journal bearings

WHIRL FREQUENCY RATIO

$$\sigma = \frac{\mu \, \Omega \, L \, R}{4W}\left(\frac{L}{c}\right)^2$$



0.50



High speed
Low load
Large viscosity

Low speed
Large load
Low viscosity

High speed
Low load
Large viscosity

At centered condition

$$k_{XX} = k_{YY} = 0; \quad c_{XX} = c_{YY}; \quad k_{XY} = -k_{YX}; \quad c_{XY} = c_{YX} = 0$$

$$k_{eq} = \left(k_{XX}\, c_{XX} + c_{XY}\, k_{XY}\right)/c_{XX} \quad \mathbf{=0}$$

Figure 18

Whirl frequency ratio

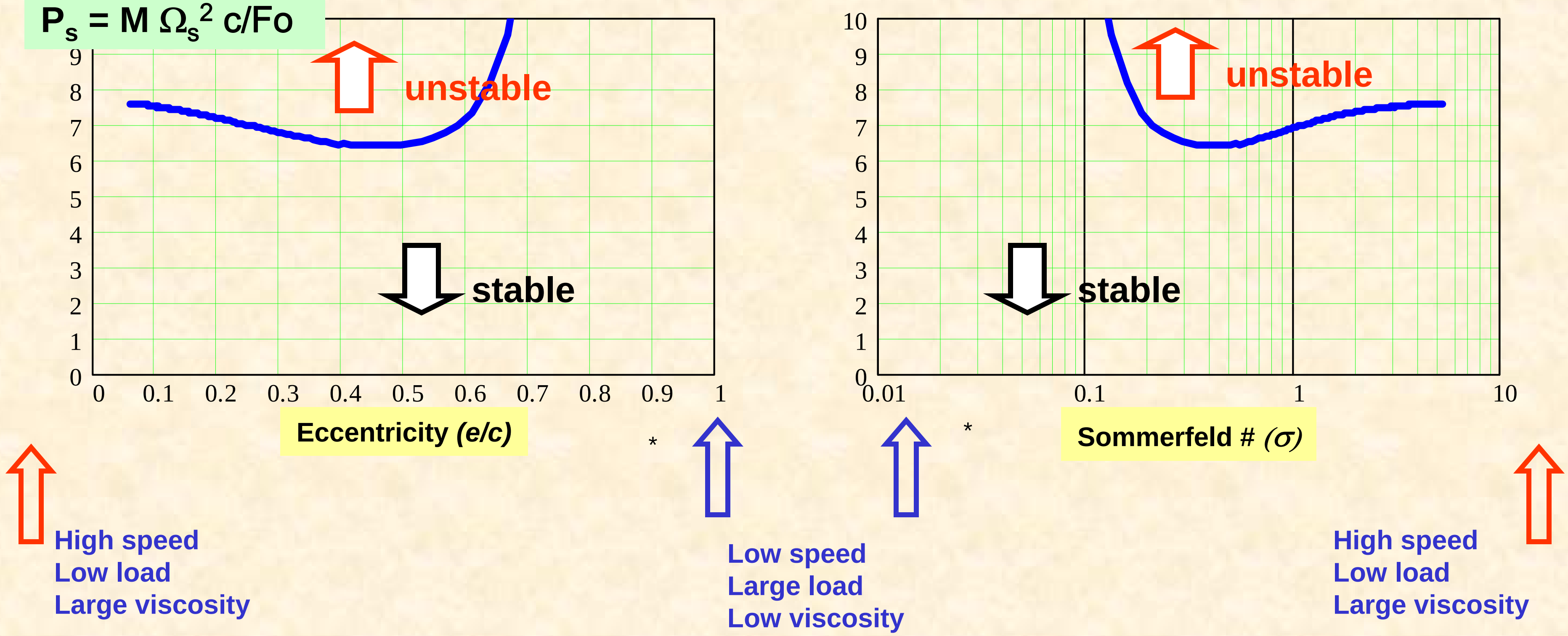
$$\frac{\omega_s}{\Omega_s} = \frac{k_{XY}}{c_{XX}} = 0.50 \quad \text{as} \quad \varepsilon \rightarrow 0$$

Rotor becomes unstable at speed = twice system natural frequency

Threshold speed of instability

$$\sigma = \frac{\mu\,\Omega\,L\,R}{4W}\left(\frac{L}{c}\right)^2$$

$P_s = M\,\Omega_s^2\,c/Fo$

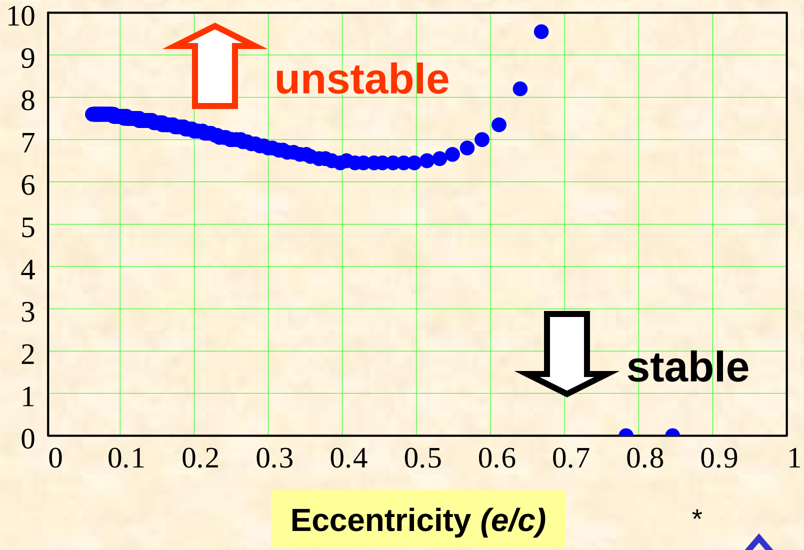


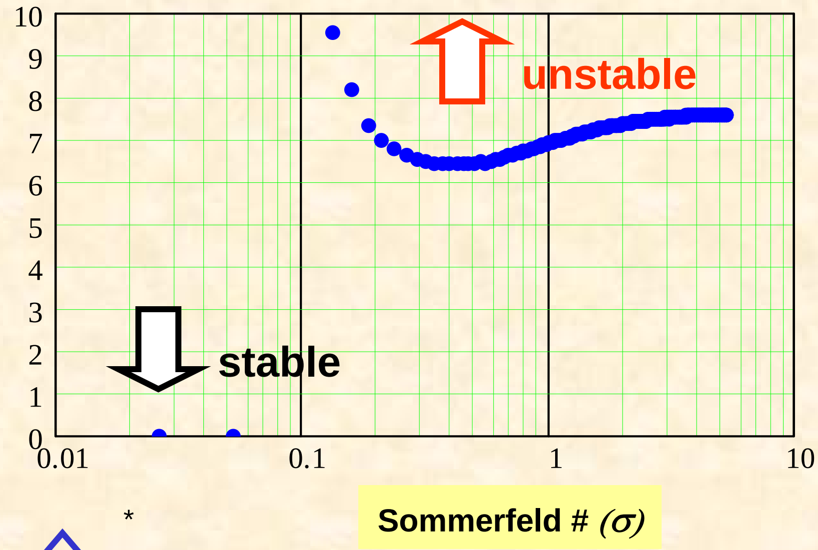
Fully stable for operation with ε > 0.75, all bearings *(L/D)*.
Threshold speed decreases as eccentricity *(e/c)* → 0

Figure 19 Threshold speed of instability versus eccentricity and design number (σ)

CRITICAL MASS

$$\sigma = \frac{\mu\,\Omega\,L\,R}{4W}\left(\frac{L}{c}\right)^2$$





High speed
Low load
Large viscosity

Low speed
Large load
Low viscosity

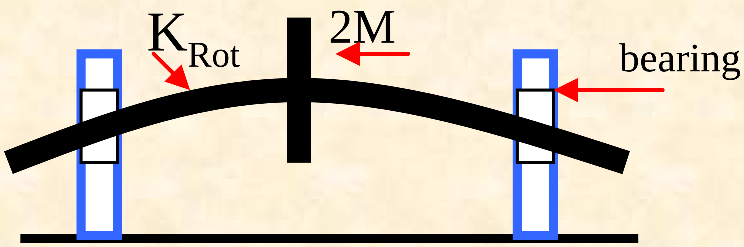
High speed
Low load
Large viscosity

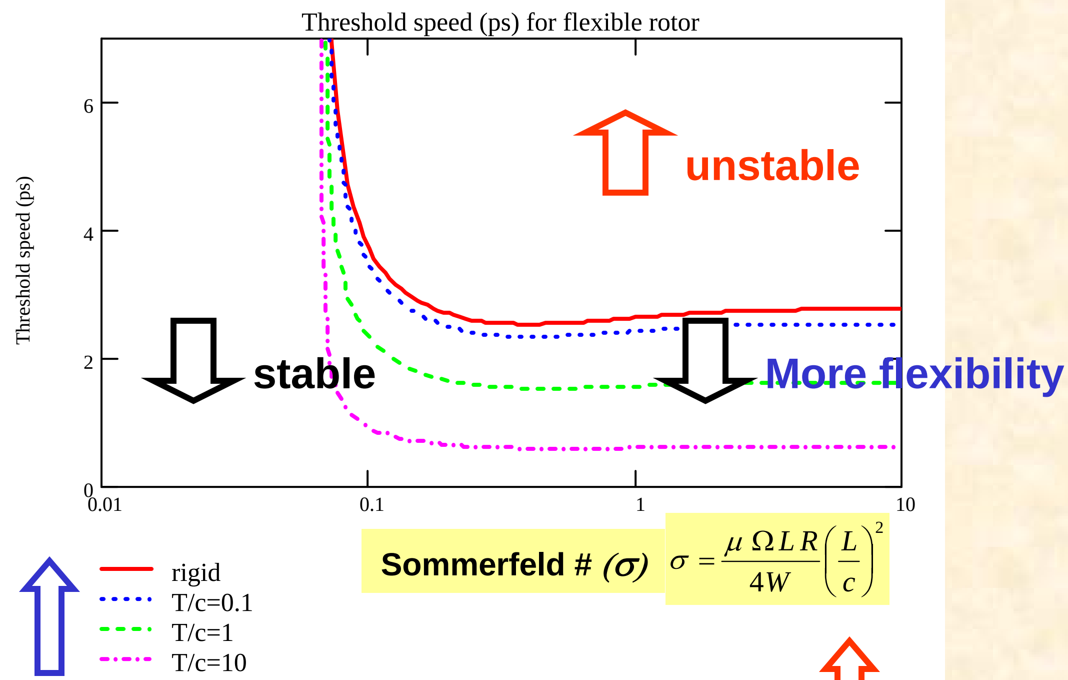
Critical mass equals maximum mass rotor is able to support stably if current operating speed = threshold speed of instability.

Critical mass decreases for centered condition. Unlimited for large (*e/c*)

Figure 20 **Critical mass versus eccentricity and design number (σ)**

EFFECTS OF ROTOR FLEXIBILITY





Sommerfeld # (σ) $\sigma = \frac{\mu \, \Omega \, L \, R}{4W}\left(\frac{L}{c}\right)^2$

$$p_{sf}^2 = \frac{p_s^2}{1 + k_{eq}\left(\frac{T}{C}\right)}$$

Static sag $T = F_o / K_{rot}$

Low speed
Large load
Low viscosity

High speed
Small load
High viscosity

Rotor flexibility decreases system natural frequency, thus lowering threshold speed of instability. WFR still = 0.50

Figure 21

PHYSICS of WHIRL MOTION

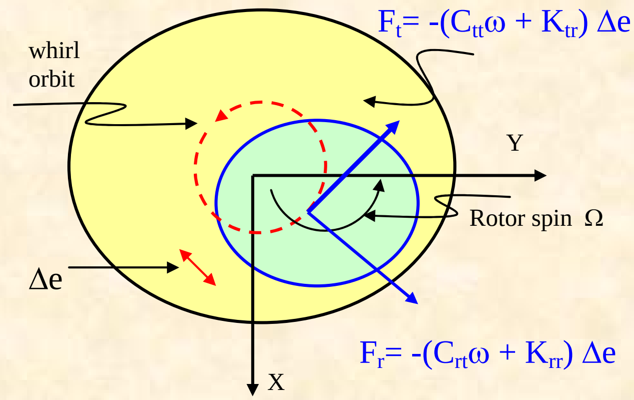
Forces in rotating coordinate system

$$\begin{pmatrix} F_r \\ F_t \end{pmatrix}_d = -\begin{bmatrix} K_{rr} & K_{rt} \\ K_{tr} & K_{tt} \end{bmatrix}\begin{pmatrix} \Delta e \\ e_0 \Delta\phi \end{pmatrix} - \begin{bmatrix} C_{rr} & C_{tr} \\ C_{tr} & C_{tt} \end{bmatrix}\begin{pmatrix} \Delta\dot{e} \\ e_0 \Delta\dot{\phi} \end{pmatrix}$$

Bearing force coefficients at *(e/c)*=0

$$K_{rr} = K_{tt} = C_{rt} = C_{tr} = 0$$

$$\overline{K} = K_{rt} = -K_{tr} = \frac{\Omega}{2}\overline{C}; \qquad \overline{C} = C_{tt} = C_{rr} = \frac{\mu R L^3}{C^3}\frac{\pi}{2}$$



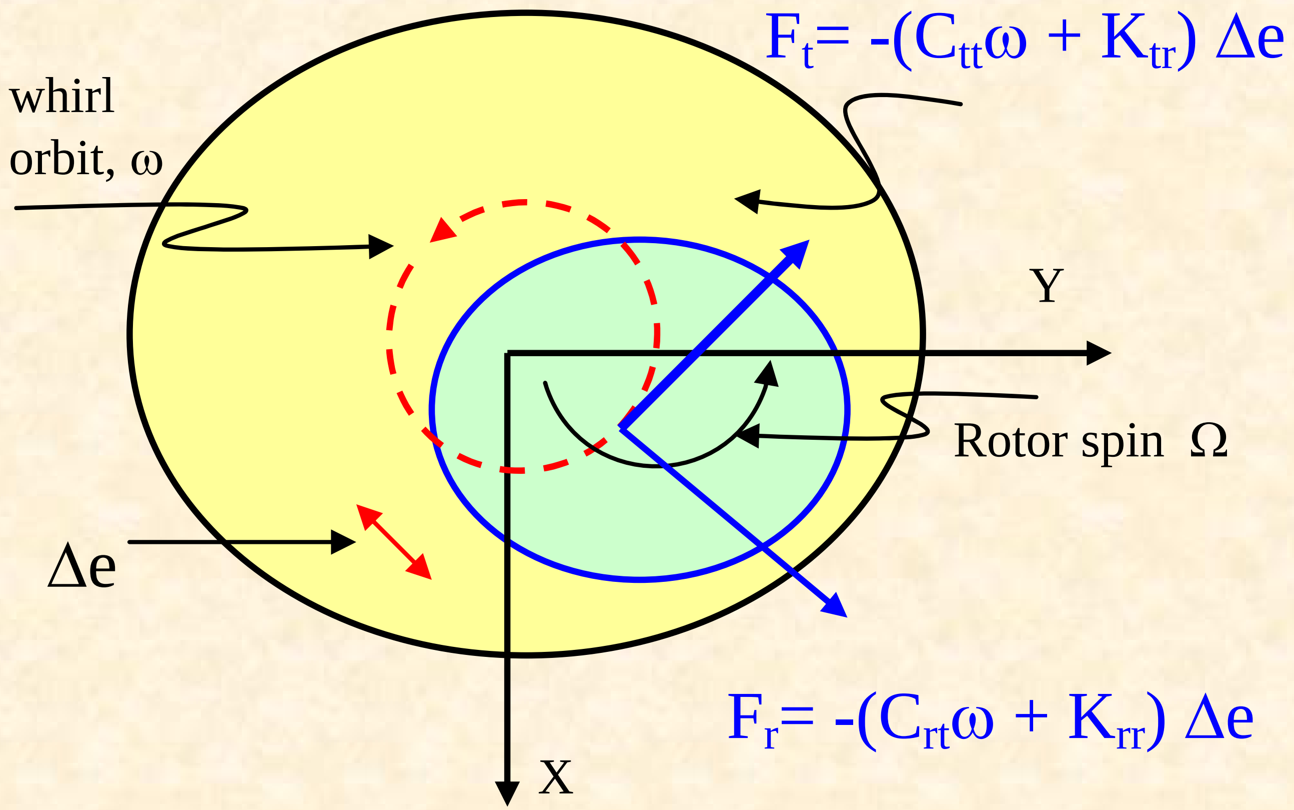
Resultant forces

$$F_{r_d} = 0; \qquad F_{t_d} = -(C_{tt}\,\omega - K_{rt})\,\Delta e$$

At centered condition: No radial support, tangential force must be < 0 to oppose whirl motion

Figure 22

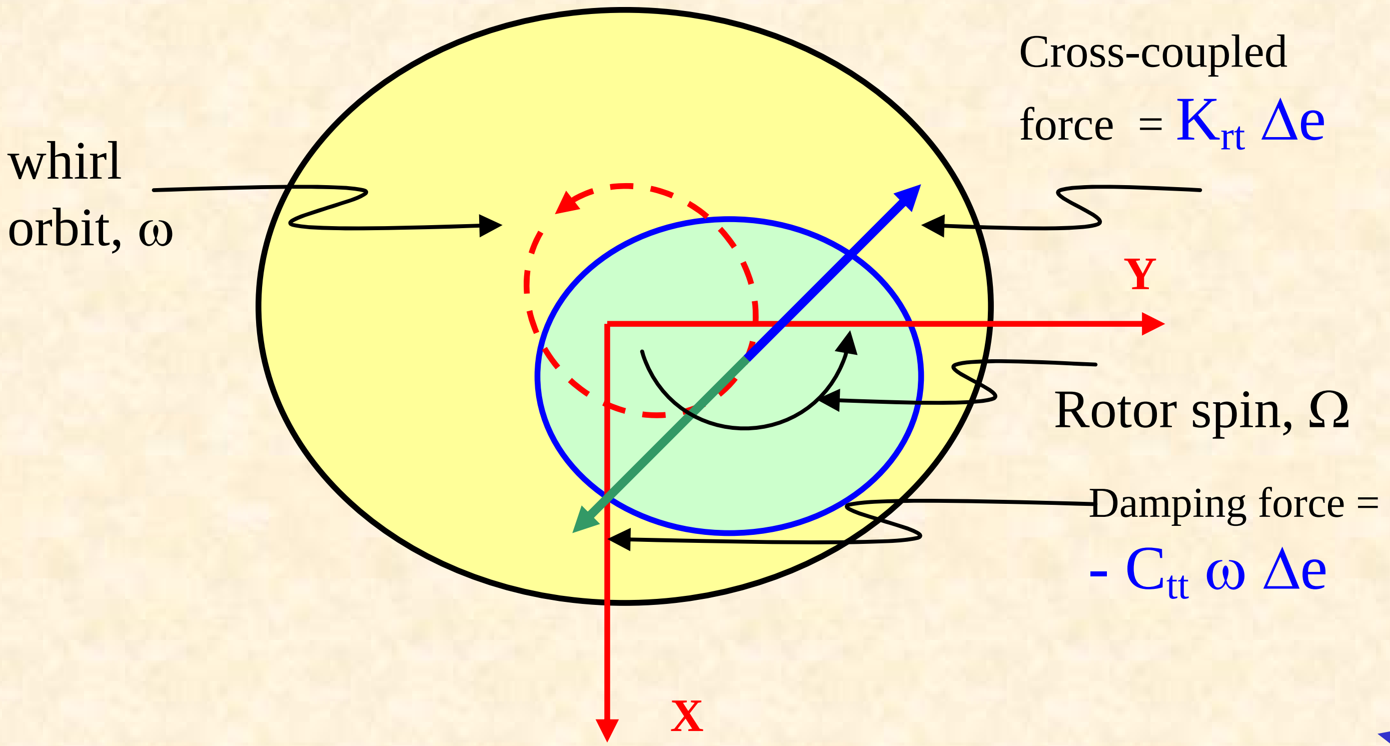
PHYSICS of WHIRL MOTION



$$\left(C_{tt} - \frac{1}{\omega}K_{rt}\right) = C_{eq} < 0$$

Loss of damping for speeds above ω_s

Figure 22 **Force diagram for circular centered whirl motions**

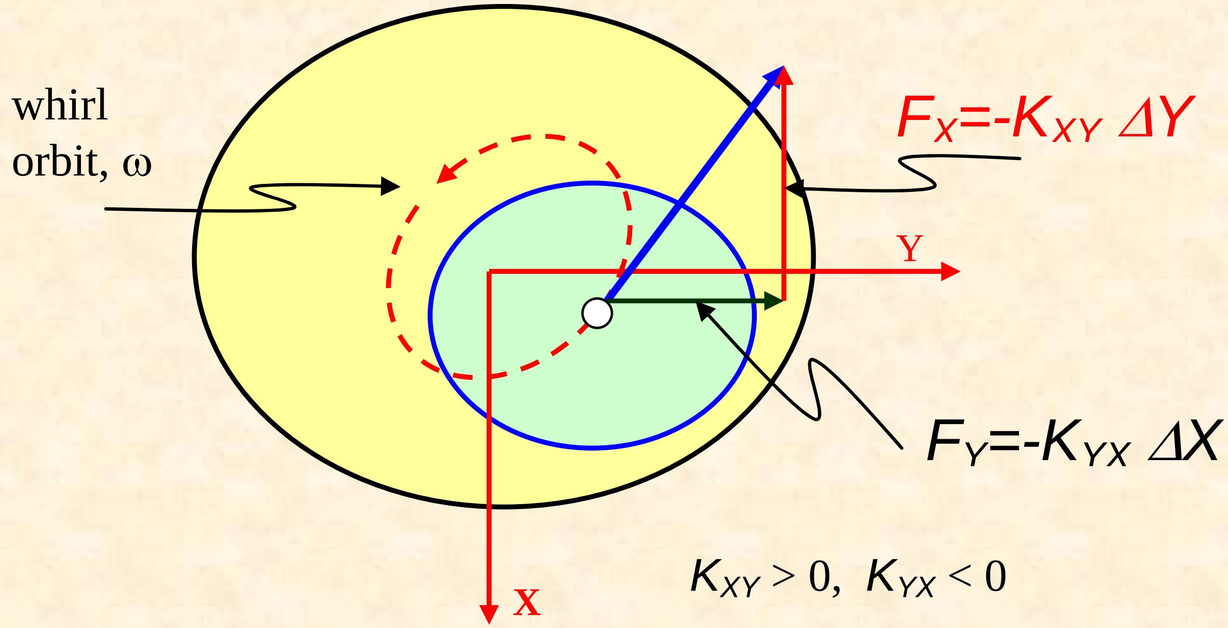


$$\left(C_{tt} - \frac{1}{\omega} K_{rt}\right) = C_{eq} < 0$$

Cross-coupled force is a FOLLOWER force

Figure 23 **Forces driving and retarding rotor whirl motion**

PHYSICS of WHIRL MOTION



$K_{XY} > 0$, $K_{YX} < 0$

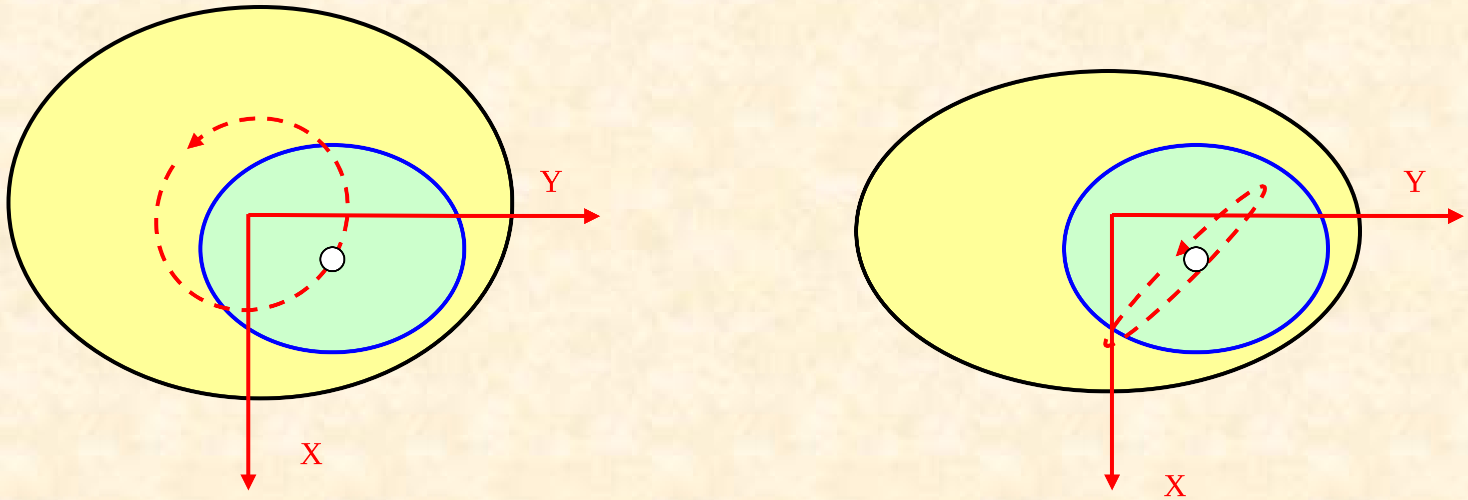
$\Delta X<0$, $\Delta Y>0$

$$E = -\left(2\pi\Delta e^2\right)\left(C_{tt}\,\omega - K_{rt}\right) = -2Area_{orbit}\,C_{eq}\,\omega$$

Work from bearing forces. E<0 is dissipative; E>0 adds energy to whirl motion

Figure 24 **Follower force from cross-coupled stiffnesses**

PHYSICS of WHIRL MOTION

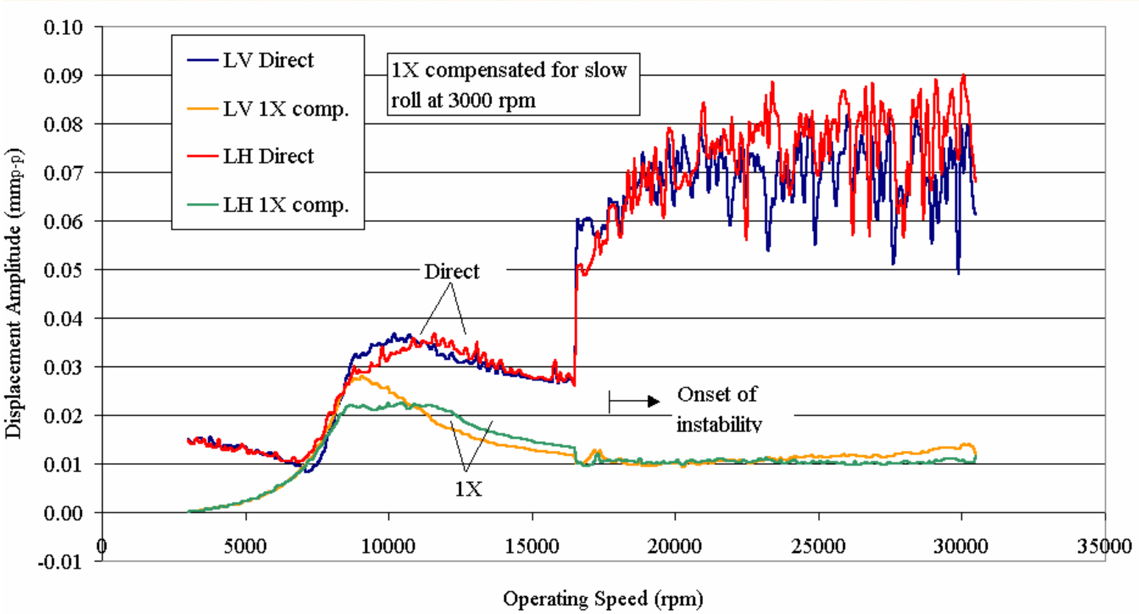


Energy from cross-coupled forces = Area (K_{xy}-K_{yx})

Bearing asymmetry creates strong stiffness asymmetry – a remedy to reduce potential for hydrodynamic instability

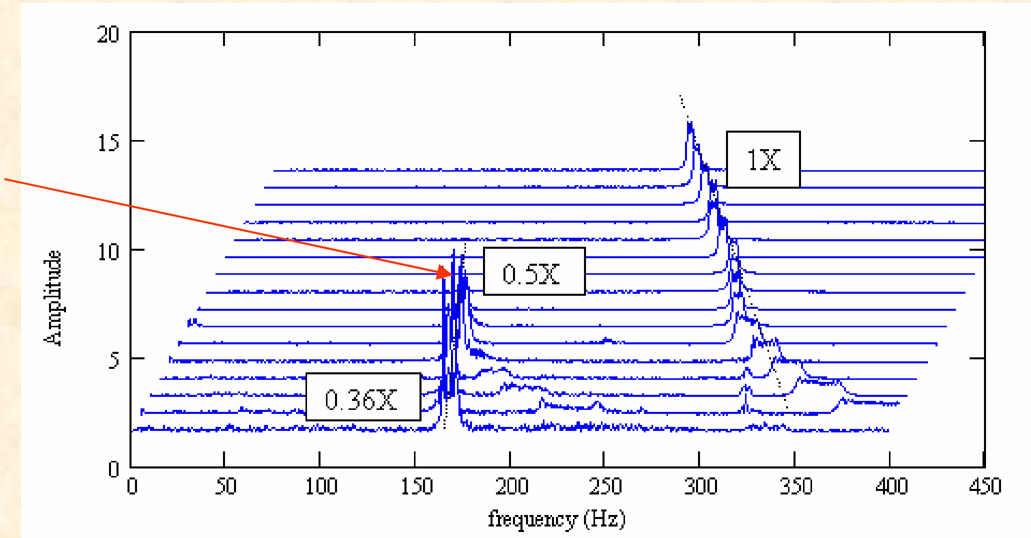
Figure 24 **Influence of bearing asymmetry on whirl orbits**

EXPERIMENTAL EVIDENCE of INSTABILITY



Amplitudes of rotor motion versus shaft speed. Experimental evidence of rotordynamic instability

Waterfall of recorded rotor motion demonstrating subsynchronous whirl

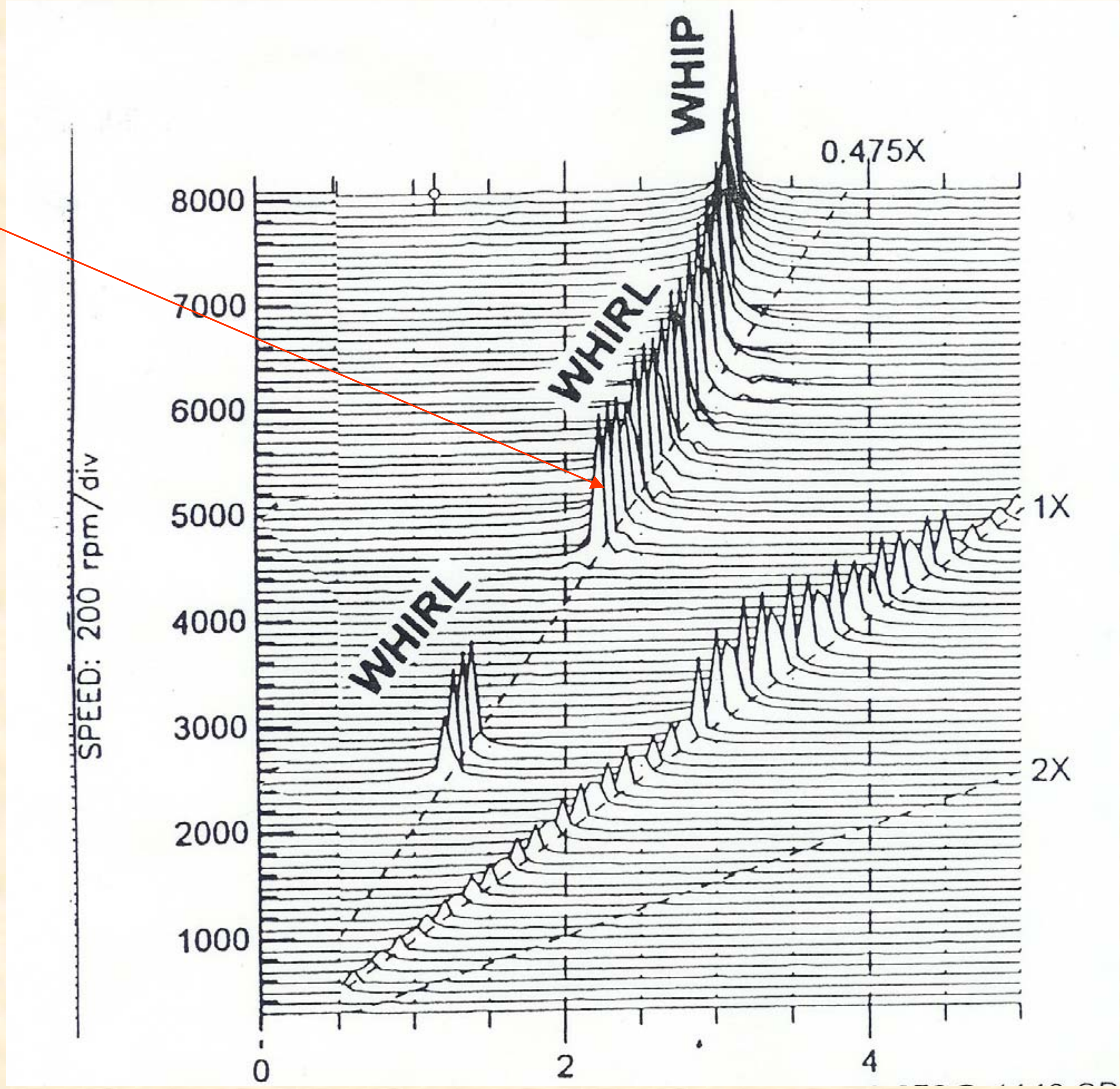


Figures 25 & 26

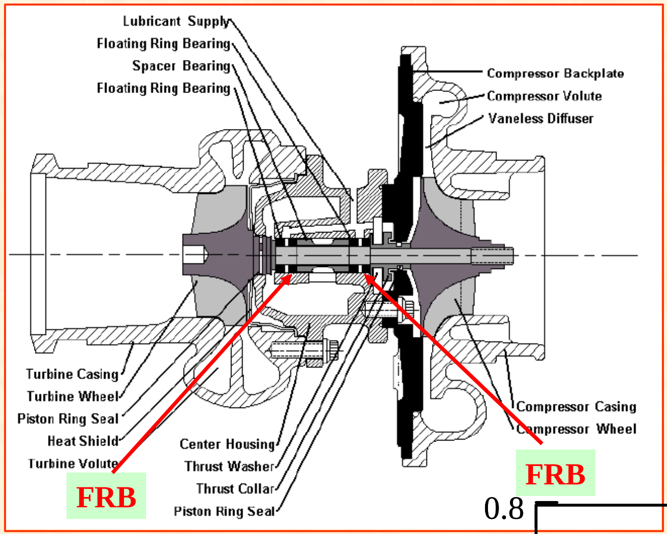
EXPERIMENTAL EVIDENCE of INSTABILITY

WFR ~ 0.47 X

Transition from oil whirl to oil whip (sub sync freq. locks at system natural frequency)

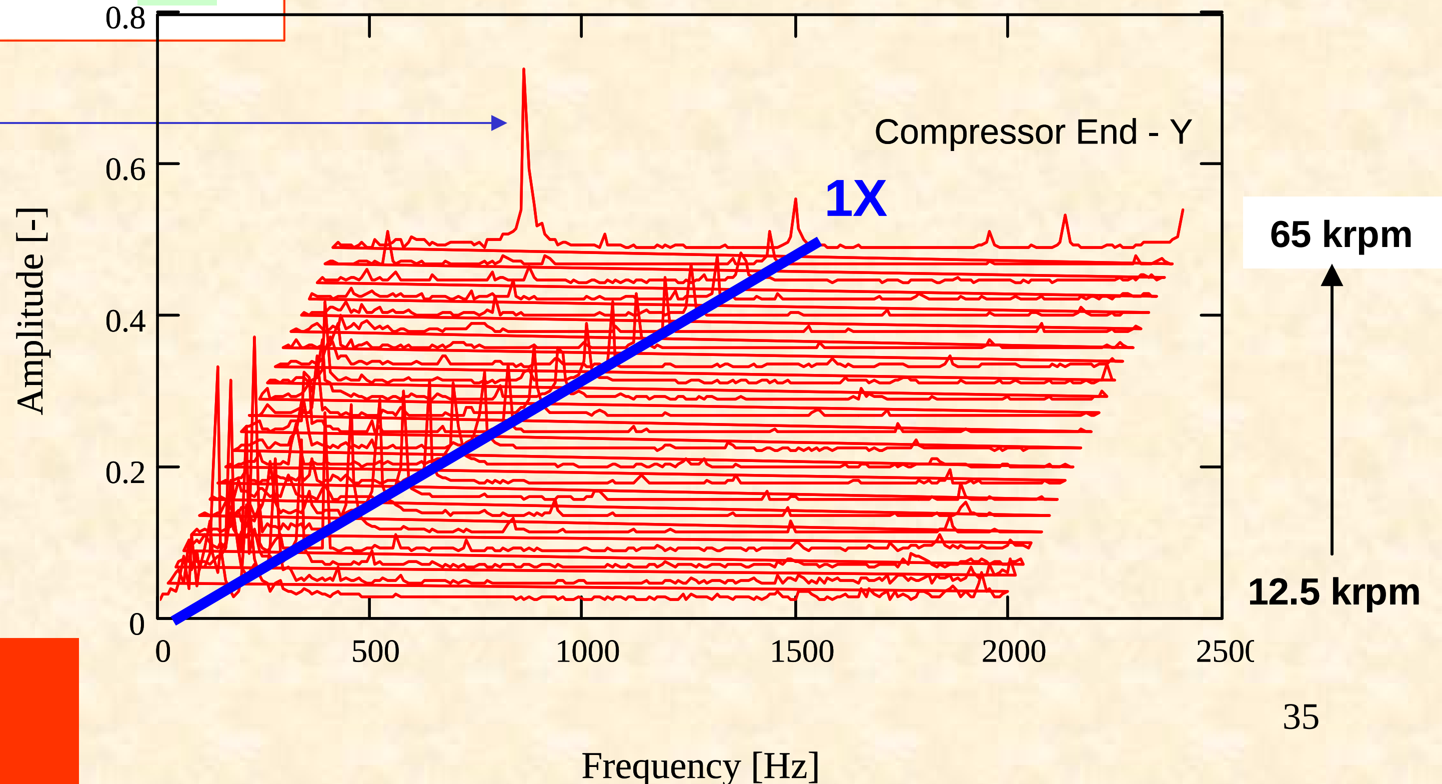


EXPERIMENTAL EVIDENCE of INSTABILITY



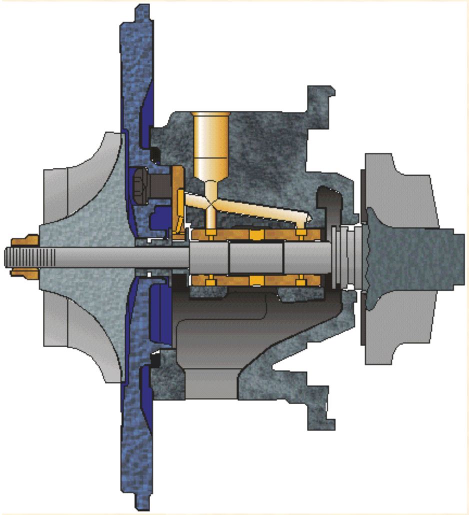
TC supported on floating ring bearings

WFR ~ 0.50 X



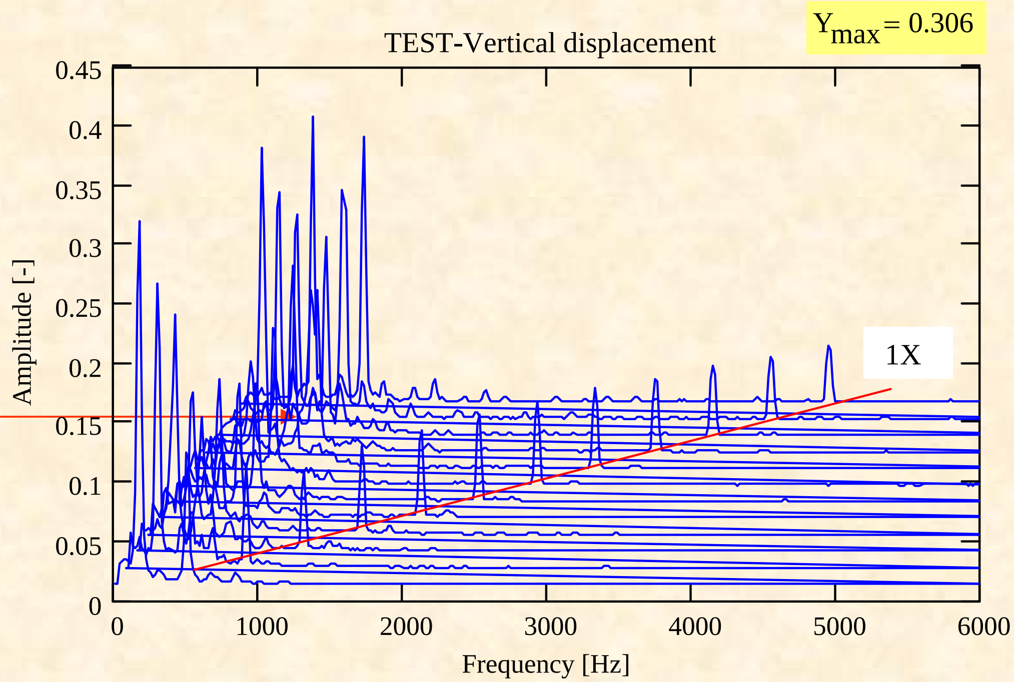
Automotive Turbocharger

EXPERIMENTAL EVIDENCE of INSTABILITY



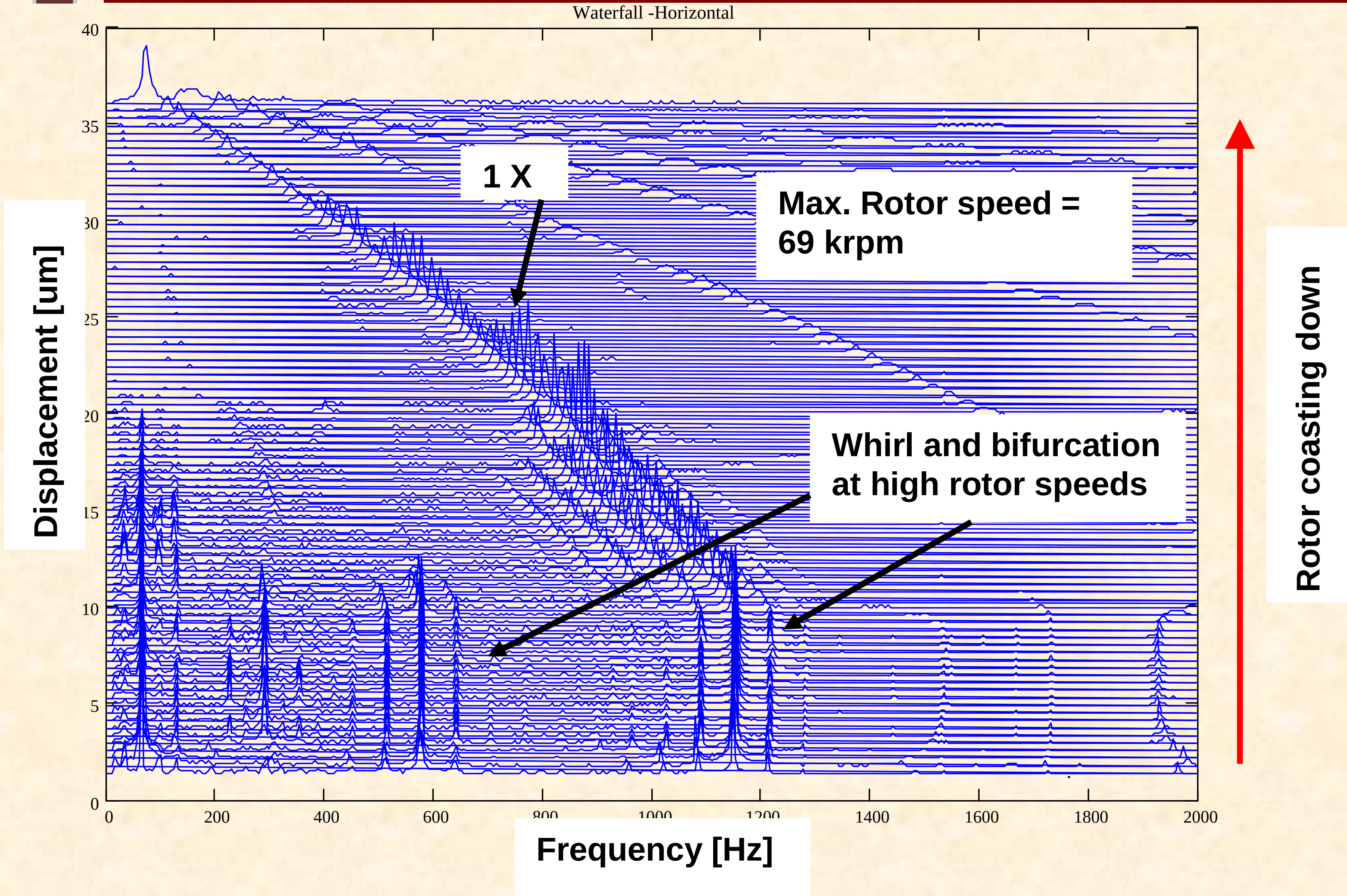
TC supported on semi-floating ring bearings

Multiple sub-synchronous motions



Automotive Turbocharger

EXPERIMENTAL EVIDENCE of INSTABILITY



Metal Mesh Gas Foil Bearing

CLOSURE

Commercial rotating machinery implements bearing configurations aiming to reduce and even eliminate the potential of hydrodynamic instability (sub synchronous whirl)

Cutting axial grooves in the bearing to supply oil flow into the lubricated surfaces generates some of these geometries.

Other bearing types have various patterns of variable clearance (preload and offset) to create a pad film thickness that has strongly converging wedge, thus generating a direct stiffness for operation even at the journal centered position.

In tilting pad bearings, each pad is able to pivot, enabling its own equilibrium position. This feature results in a strongly converging film region for each loaded pad and the near absence of cross-coupled stiffness coefficients.

OTHER BEARING GEOMETRIES

Bearing Type	Advantages	Disadvantages	Comments
Plain Journal	1. Easy to make 2. Low Cost	1. Most prone to oil whirl	Round bearings are nearly always “crushed” to make elliptical bearings
Partial Arc	1. Easy to make 2. Low Cost 3. Low horsepower loss	1. Poor vibration resistance 2. Oil supply not easily contained	Bearing used only on rather old machines
Axial Groove	1. Easy to make 2. Low Cost	1. Subject to oil whirl	Round bearings are nearly always “crushed” to make elliptical or multi-lobe
Floating Ring	1. Relatively easy to make 2. Low Cost	1. Subject to oil whirl (two whirl frequencies from inner and outer films (50% shaft speed, 50% [shaft + ring] speeds)	Used primarily on high speed turbochargers for PV and CV engines

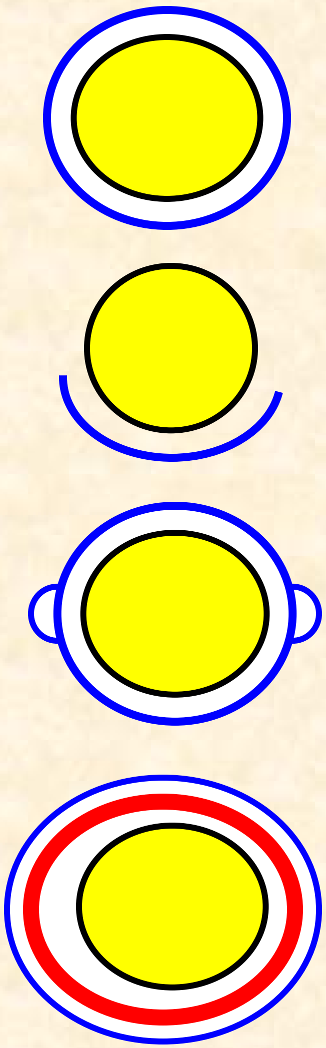
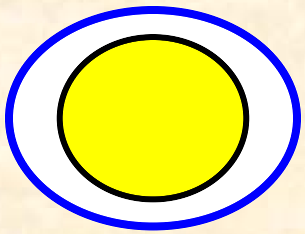
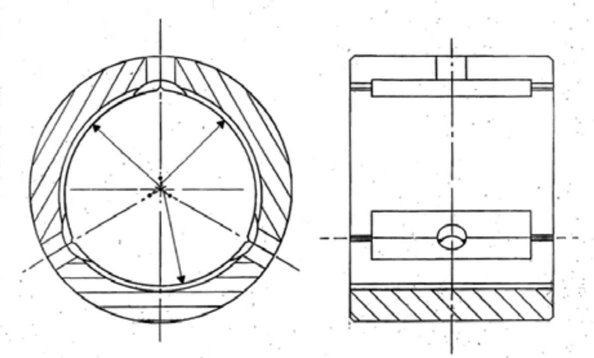


Table 2 **Fixed Pad Non-Pre Loaded Journal Bearings**

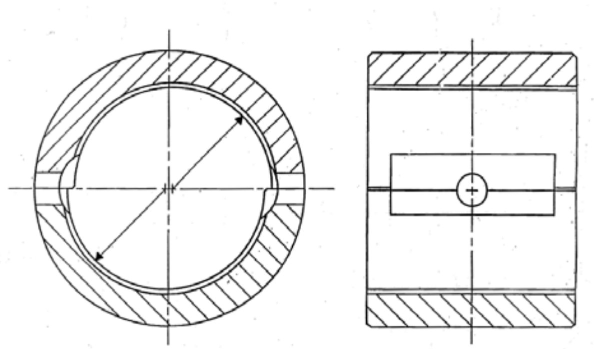
OTHER BEARING GEOMETRIES

Bearing Type	Advantages	Disadvantages	Comments
Elliptical	1. Easy to make 2. Low Cost 3. Good damping at critical speeds	1. Subject to oil whirl at high speeds 2. Load direction must be known	Probably most widely used bearing at low or moderate rotor speeds
Offset Half (With Horizontal Split)	1. Excellent suppression of whirl at high speeds 2. Low Cost 3. Easy to make	1. Fair suppression of whirl at moderate speeds 2. Load direction must be known	High horizontal stiffness and low vertical stiffness - may become popular - used outside U.S.
Three and Four Lobe	1. Good suppression of whirl 2. Overall good performance 3. Moderate cost	1. Expensive to make properly 2. Subject to whirl at high speeds	Currently used by some manufacturers as a standard bearing design

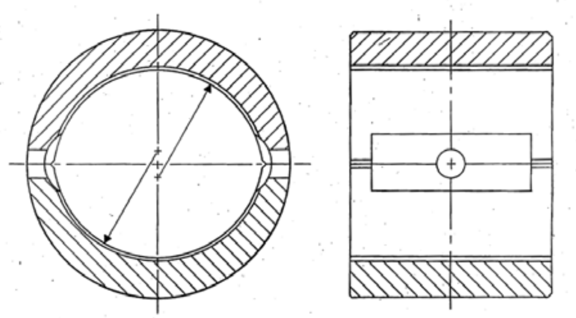




3-LOBE JOURNAL BEARING



OFFSET SPLIT BEARING



LEMON BORE BEARING

Table 2 **Fixed Pad Pre-Loaded Journal Bearings**

OTHER BEARING GEOMETRIES

Bearing Type	Advantages	Disadvantages	Comments
Pressure Dam (Single Dam)	1. Good suppression of whirl 2. Low cost 3. Good damping at critical speeds 4. Easy to make	1. Goes unstable with little warning 2. Dam may be subject to wear or build up over time 3. Load direction must be known	Very popular in the petrochemical industry. Easy to convert elliptical over to pressure dam
Multi-Dam Axial Groove or Multiple-Lobe	1. Dams are relatively easy to place in existing bearings 2. Good suppression of whirl 3. Relatively low cost 4. Good overall performance	1. Complex bearing requiring detailed analysis 2. May not suppress whirl due to non bearing causes	Used as standard design by some manufacturers
Hydrostatic	1. Good suppression of oil whirl 2. Wide range of design parameters 3. Moderate cost	1. Poor damping at critical speeds 2. Requires careful design 3. Requires high pressure lubricant supply	Generally high stiffness properties used for high precision rotors

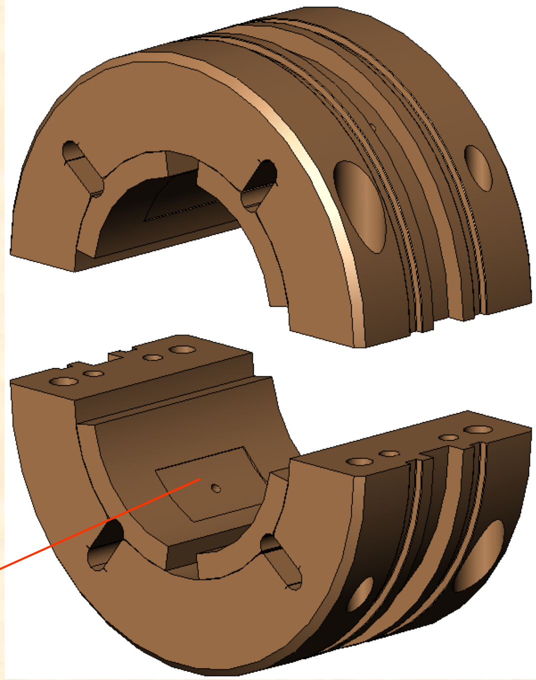
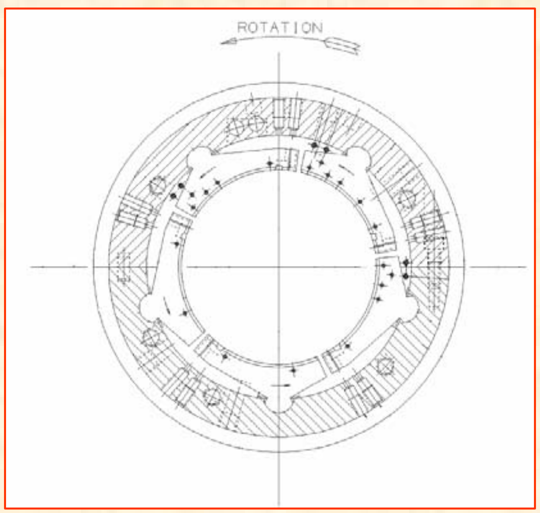
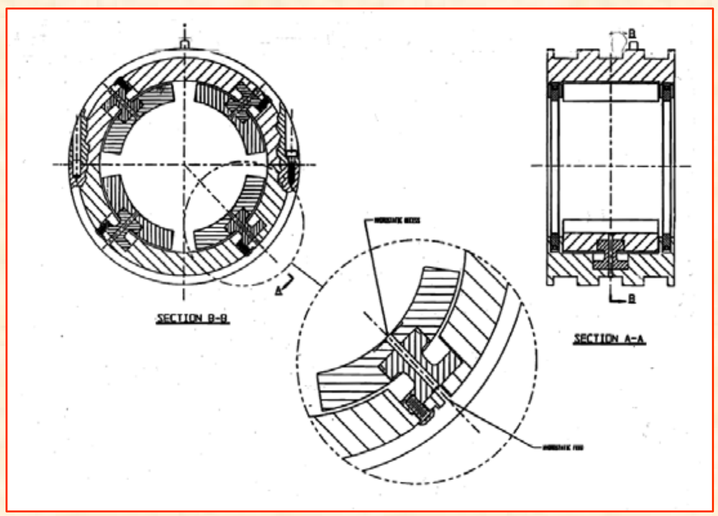


Table 2 **Fixed Pad Pre-Loaded & Hydrostatic Bearings**

OTHER BEARING GEOMETRIES

Bearing Type	Advantages	Disadvantages	Comments
Tilting Pad journal bearing **Flexure pivot, tilting pad bearing**	1. Will not cause whirl (no cross coupling)	1. High Cost 2. Requires careful design 3. Poor damping at critical speeds 4. Hard to determine actual clearances 5. Load direction must be known	Widely used bearing to stabilize machines with subsynchronous non-bearing related excitations
Foil bearing	1.Tolerance to misalignment. 2.Oil-free	1. High cost. 2. Dynamic performance not well known for heavily loaded machinery. 3. Prone to subsynchronous whirl	Used mainly for low load support on high speed machinery (APU units).





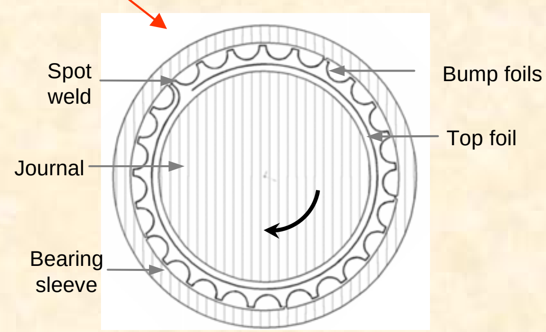


Table 3 **Tilting Pad Bearings & Foil Bearings**